PRODUCT ENGINEERING

OCTOBER
1934

•

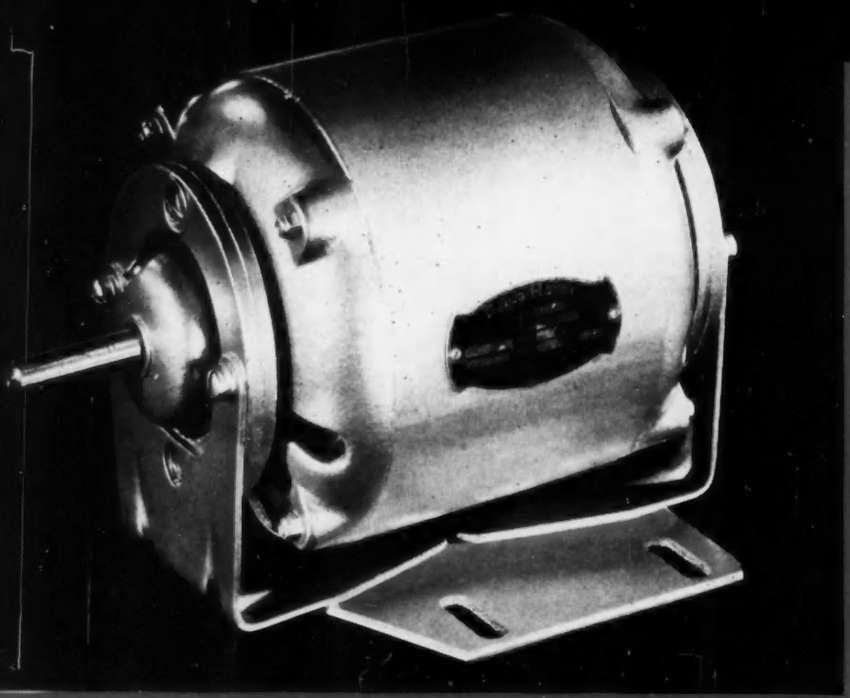
Metals—pure or alloyed, ferrous or non-ferrous, plain or ornamental—are the primary materials with which the product engineer performs his magic in this machine civilization

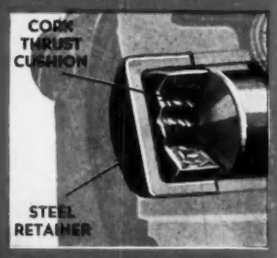
•

Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.

Delco Motors





No End-Play Noise

Sealed Lubrication

Vulcanized Rubber Cradle Mounting

These 3 FEATURES TYPIFY THE VALUE OF DELCO MOTORS

Here are three features which admirably illustrate the value that is built into Delco motors: Delco's *No End-Play* Feature—which removes the objectionable noise occasioned by end-play in the rotor shaft; Delco's *Sealed Lubrication*—which provides automatic lubrication from the moment the motor leaves the factory; Delco's *Vulcanized Rubber Mounting*—which adds a final touch of certainty to the quietness of Delco motor operation. These three features are as highly desired by manufacturers as they are definitely valuable to the ultimate consumer. They—together with Delco's reputation for quality and dependability—help to explain the preference for Delco motors among dealers and manufacturers alike.

Delco Products Corporation · Dayton, Ohio

PRODUCT ENGINEERING

VOLUME 5 NUMBER 10

KENNETH H. CONDIT, *Editor* — MASON BRITTON, *Vice-President*
GEORGE F. NORDENHOLT, *Managing Editor* — R. W. CARSON, *Assistant Editor*
LEONARD CHURCH, *Cleveland* — PAUL WOOTON, *Washington*
W. E. KENNEDY, *Manager*

November Number

With the information already piling in rapidly to the editorial desks, we can safely promise the most interesting and most valuable issue *Product Engineering* has ever had. New developments in parts, materials and finishes—how your fellow engineers are taking advantage of these new developments—types of construction used to make better products more economically—how appeal through better appearance is increasing sales—will tell the story of product engineering progress during the past year, and the present activity in product development.

McGRAW-HILL PUBLISHING CO., INC.
330 W. 42d Street, NEW YORK, N. Y.

JAMES H. McGRAW, Chairman of the Board
MALCOLM MUIR, President — MASON BRITTON, Vice-President
JAMES H. McGRAW, JR., Vice-President, Treasurer — B. R. PUTNAM, Secretary

Branch Offices: 520 North Michigan Avenue, Chicago; 883 Mission Street, San Francisco; Aldwych House, Aldwych, London, W. C. 2; Washington; Philadelphia; Cleveland; Detroit; St. Louis; Boston; Greenville, S. C.

CONTENTS for OCTOBER, 1934

SAVING MONEY BY USING COSTLY METALS 362
ROBERT WOSAK—A comparison of the design of a modern diesel engine with one using only cast iron and carbon steel.

INSTRUMENTS AND METHODS FOR RESEARCH—II 370
HANS ERNST AND MARIO MARTELLOTTI—The second of two articles on accurate instruments for design analysis.

THE ECONOMIC SELECTION OF METAL PARTS 372
LOUIS H. MORIN—Two examples that show why each type of construction and material has its advantages.

CAST IRON—THE DEAN OF FERROUS METALS 375
J. M. LESSELLS—Some important characteristics of cast iron that are too often overlooked.

POSSIBLE CAUSES FOR BEARING FAILURE 377
C. H. LEIS—Concluding a series of articles by this author on bearing materials and design.

THE SIGNIFICANCE OF WR^2 379
JOHN W. HARPER—A simple explanation of this factor and how it is calculated.

66 CAUSES OF DEFECTS IN FINISHED STEEL PARTS 382
WERNER H. C. BERG AND STANLEY P. ROCKWELL—Causes of defects and nine tests that can be used to determine the cause.

SELECTING ALLOY STEELS FOR HIGH TEMPERATURE 384
A comparison of creep strength and corrosion resistance of steel alloys.

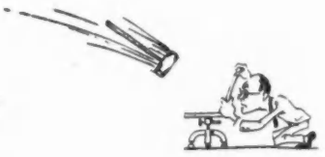
RETAINING AND LOCKING DEVICES—III 390
ADAM FREDERICKS—The third of four groups of typical designs.

TABLES FOR MOMENTS OF INERTIA 399
JOHN W. HARPER—Simple method for obtaining the inertia or radius of gyration of rotating bodies.

Editorials 361
Product Developments 366
Question and Comment 386
News 392
Meetings and Exhibitions 393
New Materials and Parts 394
Manufacturers' Publications 397
Books and Bulletins 398

Published monthly. Price 35c. a copy. Subscription rates—United States, Mexico, Central and South American countries $3 a year. Canada, including duty, $3.50 a year. All other countries $5 a year or 20 shillings. Entered as second-class matter January 17, 1930, at Post Office at New York, N. Y., under the act of March 3, 1879. Cable address "McGrawhill, New York." Member of A.B.P. Member of A.B.C. Copyright 1934 by McGraw-Hill Publishing Co., Inc. Printed in U.S.A. by the Schweinler Press, N. Y.

Intimate Correspondence



Education

PRODUCT ENGINEERING
NEW YORK

The September issue of *Product Engineering* refers to "The School of Experience," and says, "Education merely furnishes the tools." What is education as viewed by modern ideas?

"Engineering" (London), Oct. 23, 1931, page 532 refers to the late Sir James McKechnie and what he accomplished not as a B.S. or M.S. but as a great engineer. "He was apprenticed when he was fifteen and he attended Mechanics Institute evenings."

Why not refer to this in your issue as an inspiration. Great leaders are born and if they are caught when they are young they can be developed if they get the proper "Education" as Sir James did. S.P.E.E. and all they may do will never aid.

The study or drive "To Train the Mind." Is that it? "Learning by Rule and Rote" (largely Rot) is aimed to qualify What? Failure is the ultimate goal. I quote page 345, lines 10 to 13, in the September number: "A shaft designed hollow to give maximum strength with minimum material, snapped because it was too stiff to absorb the impact loads to which it was subjected." Is that an opinion or a technical analysis? It is silly, it is not even based upon common sense. Would Sir James advocate a flexible crankshaft and support in an ocean liner? He would not.

JAMES MCINTOSH
Cleveland

MR. McINTOSH'S remarks are truly Scotch in their pithiness, but he misses the point a little in the reference to the hollow shaft. The case referred to in the article in September *Product Engineering* actually happened. The shaft drove the main rolls of a big rolling mill, and the impact load was of a different order from that encountered at sea. The resilient member cured the trouble.

Clippings

PRODUCT ENGINEERING
NEW YORK

Clippers may become a pest but you went out of your way to give me a lengthy reply. I see your side; in fact I believe I had a fair idea of it before I wrote.

Don't bother to reply this time. I don't expect it.

My curiosity was aroused as to just which articles I clipped so I tabulated the following:

1—Bronze Bearing Alloys by C. H. Leis, pp. 202, 3, 4, 5 and 6, June, 1934.
2—Charts for Obtaining Spring Dimensions Directly by F. Franz, pp. 215, 6 and 7, June, 1934.
3—Determining the Critical Speeds of Shafting by W. C. Willard, pp. 221, 2 and 3, June, 1934.
4—Cams for High Speeds—I by C. N. Necklutin, pp. 250, 1, 2 and 3, July, 1934.
5—Bearings by C. H. Leis, pp. 263, 4 and 5, July, 1934.
6—Cams for High Speed—II by C. N. Necklutin, pp. 296, 7, 8 and 9, August, 1934.
7—Fundamental Requirements for Bearing Lubrication by C. H. Leis, pp. 303, 4 and 5, August, 1934.

I agree with you that in none of the above cases have I been inconvenienced by having two articles overlap. That is a very great help for us.

I can and do trim the pages. Would much rather have them so I can trim them than have them too small. The slightly smaller size for the page was just an idea for consideration when and if it was worth the trouble.

You give us more than a fair break. Features which would make your magazine 100 per cent perfect for this Clipper would make it 50 per cent more or less perfect for someone else. I don't want to run a good thing into the ground so forget it.

I will ask just one thing. May I have an extra copy of the two articles numbered 5 and 7?

M. D. HANES

IF EVERYONE adopted the reasonable attitude toward his special requirements in a technical paper which characterizes Mr. Hanes' letter our job would be a much simpler one. We really are anxious to make *Product Engineering* as practically useful as possible and we will go a long distance out of our way to meet the criticisms and grant the requests of subscribers.

A Complete File

PRODUCT ENGINEERING
NEW YORK

I am particularly interested in the type of work which you are doing on pages 310-11 of *Product Engineering*. These pages are quite helpful to us in the sense that they give us a comparative and quite thorough coverage of certain accepted standards of engineering design. We use these pages to considerable advantage and it seems to me that it is a definite contribution to anyone in the machine designing field.

I am not sure that we have had all of these pages in the past, and I am wondering if there is a file or publication where these particular pages are grouped.

RAY PERRY
Rockford

MR. PERRY'S use of *Product Engineering* reference sheets in his engineering department coincides exactly with our purpose in printing them. As stated before on this page we are always glad to try to supply missing pages in order to complete a file and we are usually able to do so if the request does not run back too many years.

A Vote

PRODUCT ENGINEERING
NEW YORK

I would like a reprint of clipping of the article "Shot-Welded Stainless Steel" part I, which appeared in July issue.

I believe you furnish such reprints to your subscribers. My subscription started with the August number, and I would like the first part of the article.

CARLTON COBERT

P.S. Merely for purposes of record or census or what have you, cast my vote for more and more articles of Industrial Control—all phases. The one on contacts in the May issue (by Seeger) was excellent.

C. S. C.

THE more votes like this one we get, the better able we are to plan issues of *Product Engineering* that will satisfy everyone.

In Order

PRODUCT ENGINEERING
NEW YORK

Thank you for the second installment of Practical Data and Formulas for the Design of Worm Gears. When I first asked for this article I did not know that it was in two sections. Will you be so kind as to send me the first installment, January, and Wire Locks and Snap Ring Fastenings, December, '33 and Gear Trains for Changeable Speed Ratios, February?

Those gear articles by Mr. Grill were so practical and complete it would be a shame not to have the series complete by including in the future data on Helical or Spiral Gears. I trust that this suggestion is not out of order for, as my two-year subscription started in March, this may have been missed.

Your consideration is certainly appreciated.

D. A. GRIER
Indianapolis

SUGGESTIONS like this one are certainly not out of order. Perhaps you are getting a bit tired of reading letters like the foregoing ones on page 2, but we are taking a chance on that because they show what readers think.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. OCTOBER 1934 VOLUME FIVE, NUMBER TEN

If It's Cheaper to Kill Them, Take a Chance

ONE OF the first things an engineer would point out in analyzing the Morro Castle disaster would be the needless danger invited by using an inflammable material when fireproof substitutes are available. There is no reason under the sun why the light alloys, stainless steels or plastics could not have been used instead of wood for the deck, furniture and decorative trim of the Morro Castle superstructure. In this age of metals it seems inexcusable to use wood where a fire hazard exists. Why take a chance?

In the *New York Times* of September 16, Russell Owen in an article "Safety at Sea" states:

"Methods of preventing fires (on ships) have been advocated by engineers for years before they were adopted. There are constant experiments with slow-burning or fire-proof materials to determine the best substance to use in cabins and public rooms. But radical changes in construction are not made overnight, for here the factor of cost enters."

NO ONE who is familiar with the history of boiler explosions, marine disasters and wooden railroad coaches burning in wrecks will take issue with Mr. Owen. Only three things can cause an industry to adopt safety measures—the safer construction may be cheaper; legislation may compel it; public preference for the safer construction may make it unprofitable not to use it, regardless of initial cost.

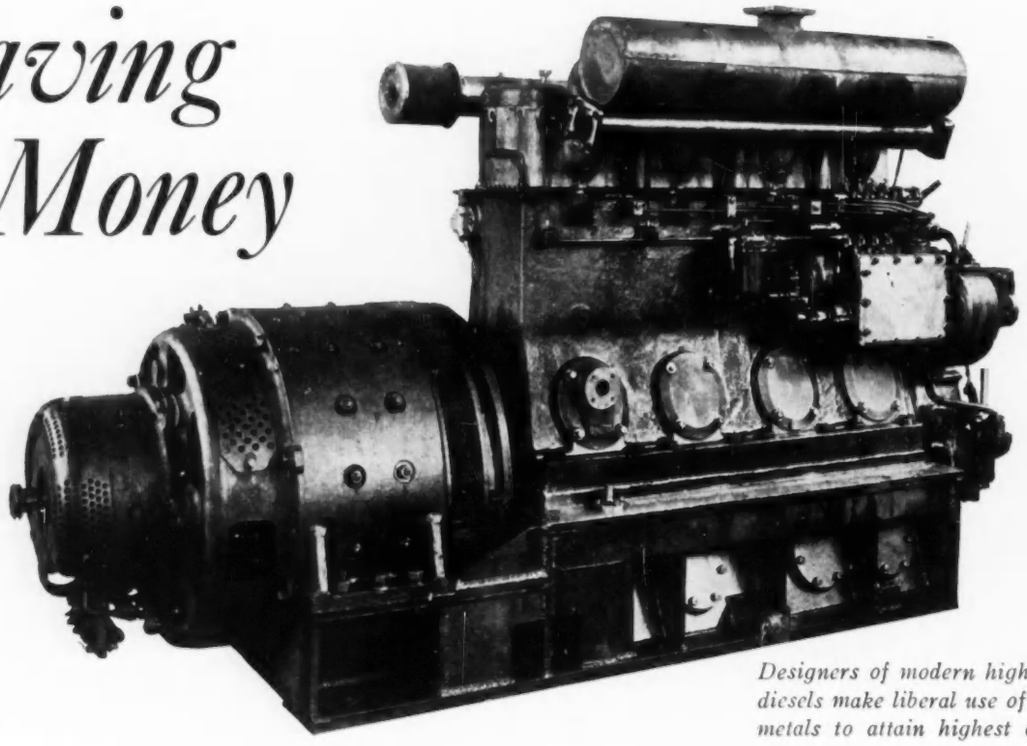
Safer construction is rarely cheaper, especially when first adopted. Almost invariably safety legislation is not enacted until there has been an appalling loss of life. Too often decisions on safety measures are made on the premise that if it is cheaper to pay for the lives that might be lost, take a chance.

It is significant that when the product is sold to and used by the "consumer," safety features in the designs follow one another in rapid succession. The automobile is the most striking example. Legislation requiring bumpers, tail lights, safety glass and many other similar items was not enacted until long after cars without these features were no longer being manufactured. The public will, expressed through the purse, is the most powerful form of compulsion.

INDIRECTLY, the same medium for compelling greater safety has operated in public utilities headed by young and aggressive men of vision who are not hampered by the dictation of short-sighted bankers whose only aim is immediate profits regardless of the cost, as long as cost is measured in anything but dollars. The airplane industry, with its rapid strides toward greater safety, is today's most striking example.

There is a powerful medium through which the deplorable situation of matching the cost of injuring and killing people against the cost of safer design can be improved. The medical profession through constant agitation has shown how an organized professional society can create a public demand that results in the protection of life and health. Our American Society of Mechanical Engineers, in its boiler code, accomplished much in the same direction. Why not extend such activities to include all machinery and structures that are purchased and operated by a company for the use of the public? Merely pointing out how a public conveyance might be made much safer would have a big effect.

Saving Money



Designers of modern high-speed diesels make liberal use of costly metals to attain highest quality

By Using Costly Metals

What would be the result if high-strength and light-weight metals were not available? Here is a comparison of a modern diesel engine with a similar one made of plain carbon steel and cast iron

ROBERT WOSAK
Oil Engine Division
Westinghouse Electric & Mfg. Company

METALLURGICAL advancement has given the designer of today superior material, such as alloy steels, alloy iron and aluminum. At the same time heat-treatments have been devised for bringing out the maximum physical characteristics of these materials. We are used to taking for granted that we have high-grade alloy steels and aluminum at our disposal. It will be interesting to consider the effect if we take away these materials and substitute plain carbon steel and cast iron.

For the purposes of illustration, we will consider a modern high-speed diesel engine designed for high-speed transportation. Such engines have to be rugged, dependable and light.

A six-cylinder engine of 9 in. bore and 12 in. stroke, runing at 900 r.p.m., and delivering 400 hp., will be taken and redesigned to use "Father's" materials. The various elements of the engine as built today will be considered, and then a comparison made with the design as it would be without the use of high-grade metal.

As built today, the power cylinder of the engine consists of a centrifugally cast nickel-chromium cast-iron cylinder liner pressed into the crankcase. For a given pressure, the wall thickness of a thin cylinder will be inversely proportional to the tensile strength of the material. With the tensile strength of nickel-chromium cast iron equal to 50,000 lb. per sq.in., and that of cast iron equal to 16,000 lb. per sq.in., the wall thickness of these liners will be increased from $\frac{3}{8}$ in. in the present engine to $1\frac{3}{16}$ in. if plain cast iron were used. The weights of these two liners would be 55 lb. and 200 lb., respectively.

Six cylinder-head studs fasten the cylinder head to the crankcase. Made of nickel-chromium steel with a

tensile strength of 120,000 lb. per sq.in., as in the present engines, these studs are 1 in. in diameter. Were carbon steel, with a tensile strength of 60,000 lb. per sq.in., used, the studs would have to be 1½ in. diameter.

As is shown above, the cylinder spacing would also have to be increased from 14¼ in. to 17⅛ in., increasing the total length of the engine approximately 14½ in. The width of the crankcase, across the top, would also be increased somewhat.

The use of plain cast-iron cylinder liners and carbon steel cylinder head bolts will necessitate an increase in the size of the cylinder head. In the modern engine this cylinder head is an aluminum casting. If cast iron were substituted, using the same wall thickness, the weight of the cylinder head will increase from 90 lb. to 235 lb., adding 7.3 per cent to the weight of the engine. This greater weight would also make it impossible for one man to handle the cylinder head during the process of manufacture or later on in service.

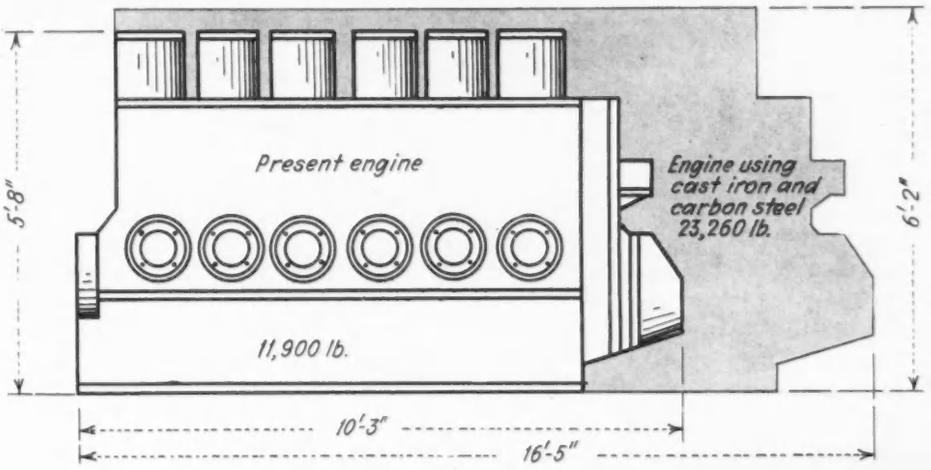
The real trouble starts when we substitute low-strength, heavy materials on reciprocating and rotating parts of the engine. Heavier moving parts have a decided influence on the loading of the bearings. Pistons on the modern engine are cast from a light, strong aluminum alloy. Changing the material from aluminum to cast iron would change the weight of each piston from 35 to 70 lb.

Changing the material of the connecting rod from chrome-nickel steel to plain carbon steel of half the tensile strength, also doubles its weight. The reciprocating parts, connecting rod and piston, would then weigh twice as much as in the modern engine.

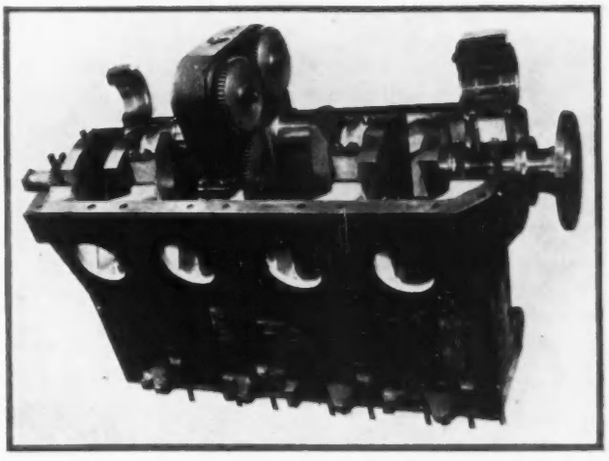
Since we started substituting, we have to stick to it also on the crankshaft. Instead of a nickel-chrome molybdenum steel with a tensile strength of 100,000 lb. per sq.in., carbon steel would have to be used. The size of the crankshaft will vary inversely as the cube root of the ratio of the strength of the materials. Thus, the present shaft diameter of 5 in. would have to be increased to 6 in. And because of the larger diameter of the crankshaft, the big end of the connecting rod would have to be redesigned to give the same stiffness. The stiffness of the connecting rod cap will vary as the cube of the distance between the supporting studs. And because the size of the rod also had to be changed due to the lower grade of material used, the weight for the same bearing length on the crankpin will be approximately one and one-half times the weight of the present rod end. This weight will have to be multiplied by the ratio of the lengths of the two bearings.

The reciprocating parts of the present engine contribute approximately 20 per cent of the mean load upon the crank pin bearing. The centrifugal force of the rotating parts of the connecting rod makes up 60 per cent of the mean bearing load.

We are assuming that the speed of the engine remains the same, therefore the forces resulting from the additional weight will be directly proportional to the forces due to the present weight of the rotating and reciprocating parts. With the weight of the reciprocating parts doubled, 20 per cent will be added to the load on the bearing. To keep the load per square inch of bearing area within the same space limits, the big end of the bearing of the connecting rod will have to be lengthened. The additional length of the bearing will make the big end of the rod still heavier, and since the centrifugal forces constitute the major part of the load, this will materially influence the bearing load. As mentioned before, the big end of the connecting rod will be about one and one-half times as heavy for the same width. To get the happy medium between load,



The shadow outline shows how much longer, higher and heavier the engine would be if made of cast iron and plain carbon steel



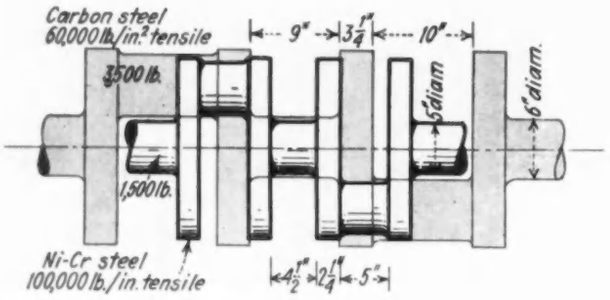
Crankcase and crankshaft assembly of the modern high-speed diesel engine, with bearing caps removed

bearing length and the weight of the rod, the length of the crankpin bearing has to be doubled. The new bearing area will then be 2.3 times that of the modern engine.

Carbon Rods Complicate Assembly

But further complications will follow. Much will depend upon the method of design of the carbon steel connecting rod. A complete change of the construction will be necessary. On the present engine the connecting rod, with the piston, can be withdrawn from the top through the cylinder after removal of the connecting rod cap. The cap can be reached through the handhole in the side of the crankcase.

But with an ordinary carbon steel rod, the increased size of the big end will not permit the removal of the



A plain carbon steel crankshaft instead of one of Ni-Cr steel, would have to be 74½ in. longer and would weigh 2,000 lb. more

rod through the cylinder. The rod would have to be split to allow for the separation of the shank of the big end, or the piston and rod would have to be withdrawn from the lower end of the cylinder, dropping it and removing the whole assembly of the piston and rod through the opening in the side of the crankcase. In either design, it will necessitate an increase in the height of the engine, because on the modern engine the piston comes within ½ in. of the webs of the crankshaft. The combined results of the "redesign" of the connecting rod is an increase in height of about 6 in. and an increase of 30 in. in the length of the engine.

To make matters worse the load on the main bearings will increase in proportion to the additional rotating weight of the crankpin and the connecting rod. Part of the rotating masses can be counterbalanced, but space will only permit a partial counterbalancing. Complete balance demands such a large counterweight that it could not be accommodated in the space available.

The main bearings have to be increased to about twice the present length. The total additional increase in engine length is 32½ in.

The crankshaft webs also have to be strengthened by increasing the thickness from 2¼ in. to 3¼ in. which, for the shaft of a six-cylinder engine adds another 12 in. to the length. The sum of all these increases will add 74½ in. to the length of the engine.

The weight of the larger crankshaft will be 3,500 lb., as compared to 1,500 lb. for the present crankshaft, adding 17 per cent to the weight of the engine.

There are two major parts left that will have to be considered—the crankcase and the bedplate.

The walls of the crankcase will have to be thickened when we substitute cast iron for the cast steel, because all the gas loads from the cylinder head to the main bearings are carried through the walls in tension.

The crankcase of the present engine weighs 2,800 lb., it is 35⅜ in. high and 92 in. long. From past experience it has been found that the weight of such a structure varies directly as the wall thickness and its outside dimensions.

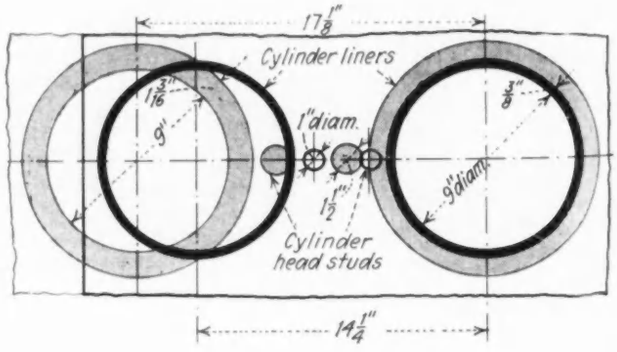
The combination of strength and stiffness are very important; strength to carry the loads and stiffness to give rigid unvarying support for the main bearings. In many cases the gas loads are transmitted through long tie rods or studs to the main bearings. Such a construction may have to be resorted to when we use cast iron. The increased length of the crankcase will demand a stiffening against deflection. Some engine builders have paid well for the extreme lightening of the engine structure. Some cases of cracked crankcases, main bearings and even crankshaft failures could be traced to the lack of rigidity of the crankcase.

Materials for the Crankcase

On the basis of an increased wall thickness from ⅜ to ½ in. and the increased length and height, the weight changes from 2,800 lb. to 7,900 lb. This increase of 5,100 lb., is tremendous.

The crankcase being the heaviest part gives the designer a good chance to compare the application of different materials on the basis of weight, price and the characteristics of the materials. Let us take a crankcase of the same outside dimension and of the same construction and just change the wall thickness to take care of the different strengths of the materials and make it out of cast iron, cast steel, welded steel and aluminum. The main drawback of cast iron is its low tensile strength which necessitates heavy walls. The advantage is the low price and the possibility of a sound casting, since the metal flows easily in the mold. A disadvantage from the point of design, and sometimes repair, is the fact that it is hard to weld satisfactorily.

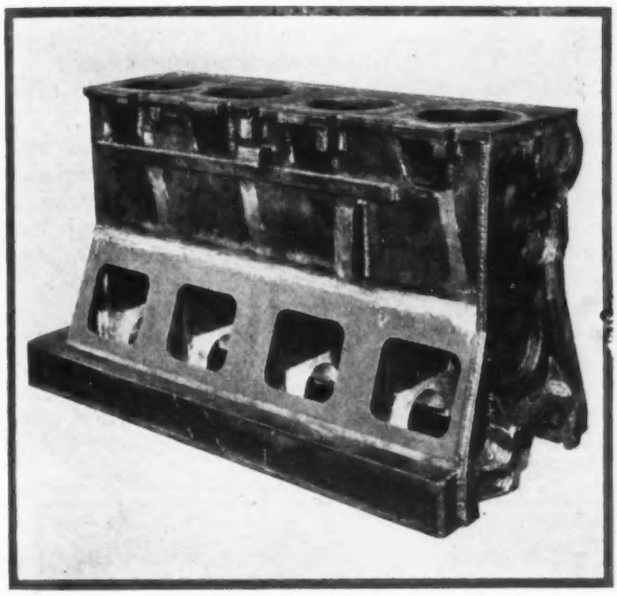
Cast steel has many advantages. It permits thin wall sections due to its high tensile strength. The wall thickness is only limited by foundry practice, since cast



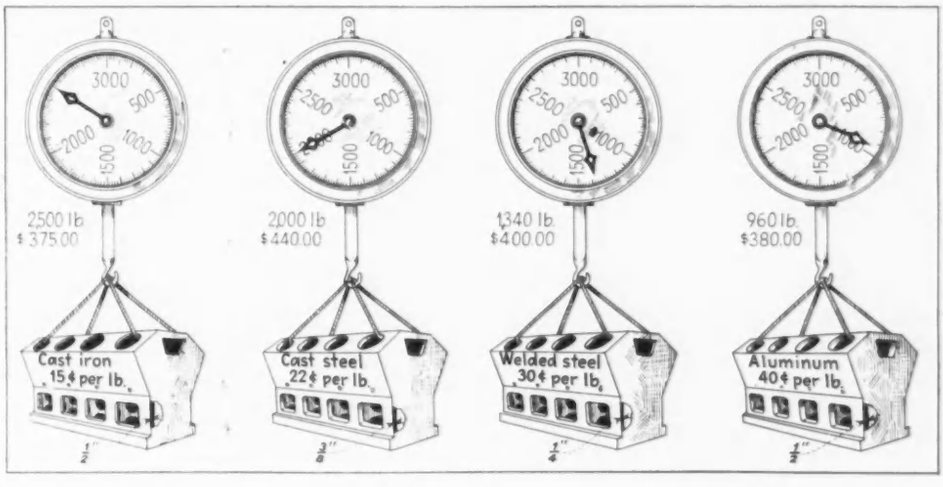
If cylinder liners were plain cast iron and cylinder head bolts plain carbon steel, 14½ in. would be added to the length of the engine

steel does not flow easily in a mold. Walls thinner than $\frac{1}{8}$ in. in a fair-sized casting are not practical. The material lends itself to a combination of welded and cast structure.

Passages for water and oil can be welded to the main structure, and, therefore, give the designer a chance to use his ingenuity and eliminate costly and often ugly piping. Repair of blow holes and cavities in the casting by welding is easy. Many a casting has been saved by this method.



Welded crankcases have proved economical and satisfactory if designed properly and annealed to relieve internal stresses and to refine the grain structure of the material adjacent to the welds



Cost of the material per pound is not a measure of the relative cost of the finished part, as illustrated by these crankcases

The drawbacks in using cast steel are casting difficulties and the higher price of the material.

A welded structure has the advantage of using highest strength materials of absolute soundness. No pattern work is involved and there is considerable latitude in the design, since for complicated sections cast steel parts can be welded right into the structure. Lightness of the structure can be easily obtained. The difficulties in using a welded structure must be considered very carefully. It requires considerable experience on the part of the designer and the welder to determine the size of the welds, the manner of assembly of the individual parts and the placing of the actual weld in the right way to eliminate deformation during welding.

Caution must be exercised in the design of welded structures because of the necessity of annealing the welded part to relieve the internal stresses set up due to localized heating during welding. The structure must be annealed also to refine the grain structure in the metal adjacent to the weld. Welded underframes, bedplates and similar parts have proved simple and satisfactory as they involve a minimum amount of welding. But for the design of complicated parts considerable development is still necessary. The cost of a welded structure is dependent upon the simplicity of the construction and the amount of welding involved.

Aluminum has the chief advantage of lightness; in strength it equals or is better than good cast iron. The disadvantages are the higher price of the material and the fact that it cannot be satisfactorily welded. Even at a much higher price, it may be the material to be used, since the difference in weight may make up for the higher cost per pound.

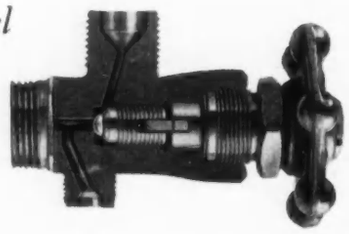
To continue with the redesign of the engine, we have to lengthen the bedplate. The weight again varies directly as the length, changing from 3,000 lb. to 4,800 lb., adding 1,800 lb., to the weight. Many smaller parts have not been considered in this discussion.

Summing it up, the redesigned engine weighs 13,260 lb., more than the present engine, which weighs 11,900 lb., making the "low-grade metal" engine approximately twice as heavy. The accompanying pictures show the comparative weights and sizes of the present engine and one designed for equal horsepower rating and speed but to be built entirely of ordinary cast iron and low-carbon steel.

The moral of the story is that it does not pay to use so-called cheaper materials on the design of a diesel engine or other high-grade machinery. It may pay to investigate the use of alloys as to their influence upon the size, weight and price of a structure. In many cases considerable savings can be made by using the best material that can be obtained, even if it costs five times as much per pound as the "cheap material that father had to use."

PRODUCT DEVELOPMENTS

Hardened Stainless Steel Used for Valve



Air Reduction Sales Company has now equipped nearly all of its oxygen cylinders with valves having a stainless steel ball seat, as shown in the accompanying picture. Although the valve proper closely resembles the old type valve in appearance, it has a hardened stainless steel ball for the valve.

It is claimed that this new design of valve not only opens and closes easily but also possesses superior sealing qualities and durability because the valve does not corrode or score in service.

Stainless steel ball valves have been used on all Airco-DB welding and cutting torches for more than three years. Their use over this period of time has demonstrated that torches so equipped rarely require valve repairs.

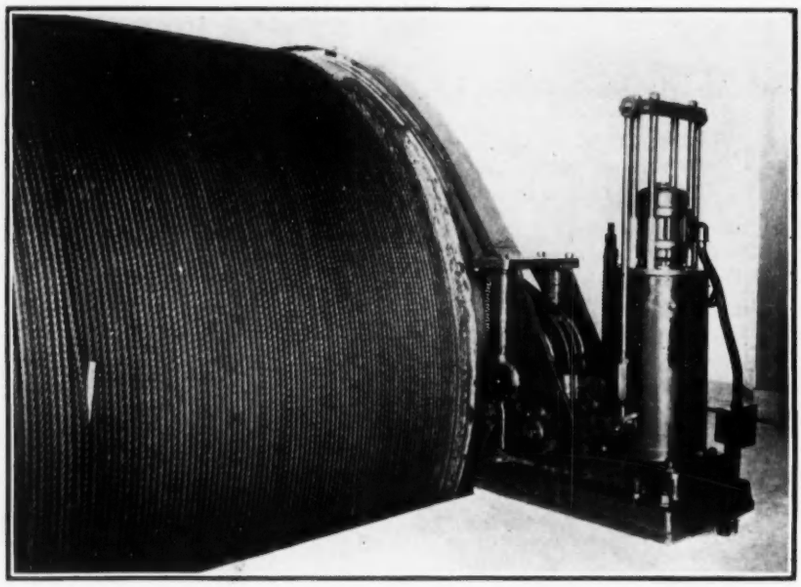
• *Cyclops Camera*

For aerial photography, clear, fast pictures are the first requirement, calling for a high grade lens and shutter. Light weight, of course, is also needed. Former designs developed by the Fairchild Aerial Camera Corp. met these requirements, but the cost of the camera made it a luxury item. A redesign, resulting in the camera shown in the accompanying illustration, greatly reduced the manufacturing cost without changing the performance, making possible a much wider appeal with a corresponding increase in sales.

Improved appearance and lower cost resulted from a careful redesign of the Cyclops camera

In the new design, the camera body is a single aluminum alloy casting, having but two machined surfaces. One surface takes the lens assembly, while the other holds the film pack and shutter. By specifying a smooth outside surface on the casting, no further machining or surface preparation is needed before applying the crackle-finish enamel.

In styling the camera, the conventional conical shape was replaced by a cylindrical design having a variety of surface contours joined by fillets of large radii. Four strips of extruded aluminum molding, chromium plated, serve only as decorations, breaking the monotony of the solid black body. Chromium plating is also used for its decorative value on the rim, name plate and in addition, on the view finder.



Application of Thrustor operated brakes on a Lidgerwood 150-ton cable drum. The Thrustor is rated at 3,200 lb. with 10 in. stroke, the brake drum being 13 ft. 1½ in. in diameter and 12 in. face width

• *Welded Diesel Cylinders*

An experimental marine diesel engine, being developed by F. B. Stearns Company, has been designed with arc welded cylinders in place of castings. The engine is 320 hp., running at 1,300 r.p.m.

Both the design and fabrication

of the cylinders involved many problems. About sixteen pieces are required for each cylinder and the tolerances on the finished work are exceedingly small. The inner sleeve is of case hardened steel and the cylinder jacket is of mild steel.

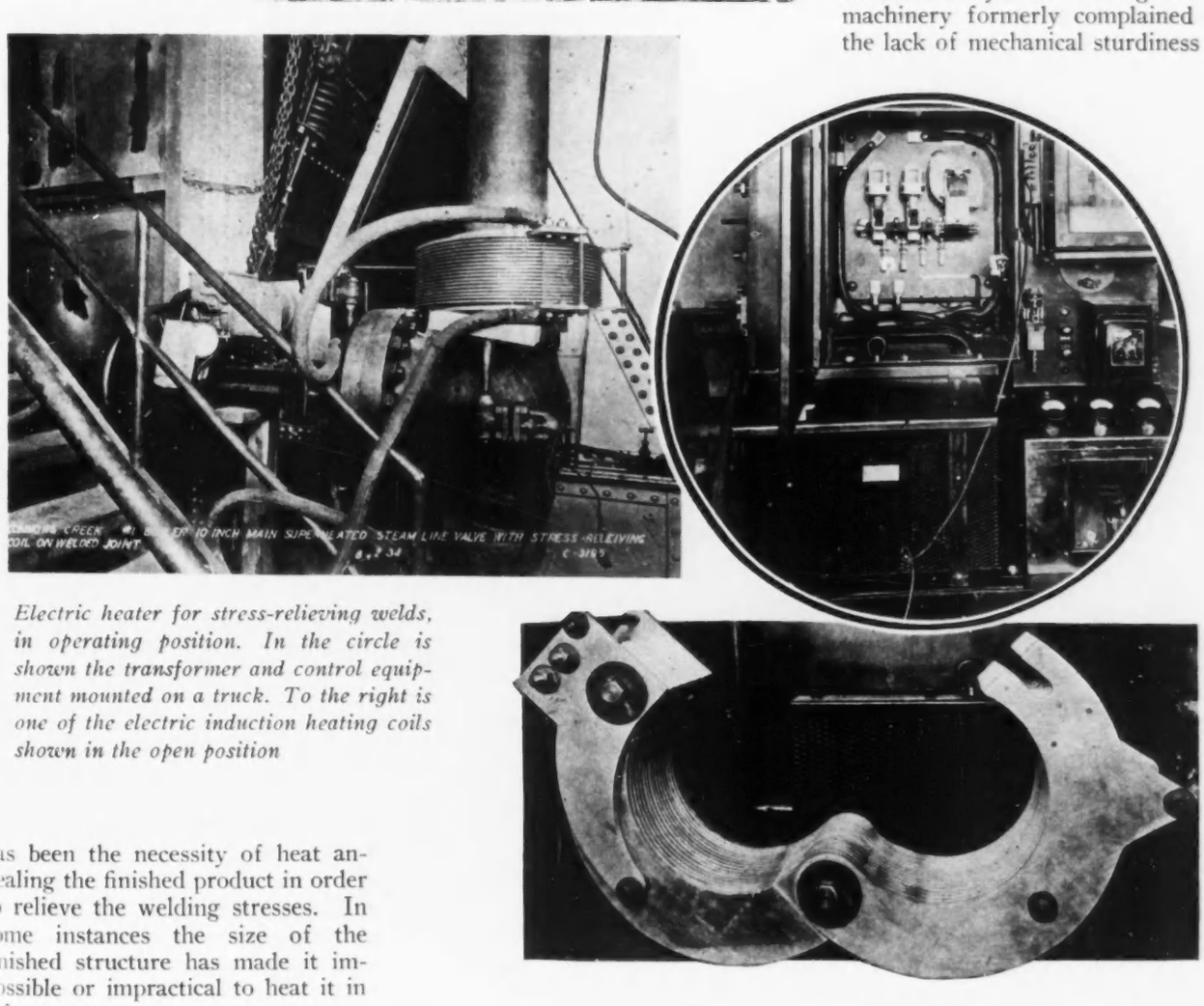
• *Weld-Stress Reliever*

One of the major problems in the design of welded structures that will be subjected to appreciable loads, has been the necessity of heat annealing the finished product in order to relieve the welding stresses. In some instances the size of the finished structure has made it impossible or impractical to heat it in a furnace.

One step toward overcoming this possible limitation in applying welding to large structures is the development of a weld-stress reliever for pipe welds. This machine, developed by the Detroit Electric Furnace Company, can be used for pipes ranging from 6 in. to 24 in. in diameter.

Although especially designed to be useful for field operation, the equipment has been designed for normalizing welded joints in any kind of pressure piping system.

Jointed rings of various sizes are provided to fit pipes within the range of diameters for which the machine is designed. Power is supplied to a special transformer which, with the necessary control equipment, is mounted on a truck for ease of transportation. In the operation of this machine, heat, generated on the induction principle, is applied to the areas to be stress-relieved, the metal being heated to a predetermined value. Speed, cleanliness and precise control are claimed as being the outstanding features of this equipment.



Electric heater for stress-relieving welds, in operating position. In the circle is shown the transformer and control equipment mounted on a truck. To the right is one of the electric induction heating coils shown in the open position

• *Stainless and Aluminum for Bicycles*

From England comes the report that Messrs. Dunford and Elliott of Sheffield are now manufacturing pedal bicycles wherein stainless steel is largely employed. It is also reported that in America one company is developing bicycle frames to be made of an aluminum alloy.

• *Improved Electric Controls*

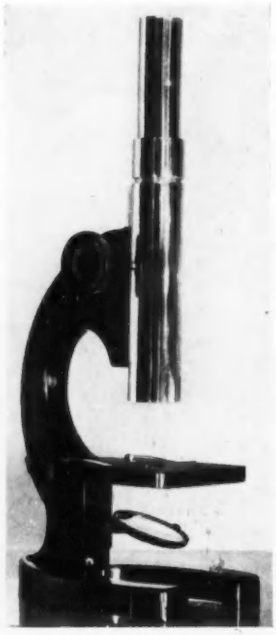
Redesign of motor control equipment, limit switches, solenoids, magnetic brakes and clutches has resulted in many new developments in the machinery field. Designers of machinery formerly complained of the lack of mechanical sturdiness in motor control devices. Apparently this complaint has been met by the electrical equipment manufacturers.

Among other things, it is reported that some machine tool builders are developing designs analogous to that of the Monarch lathe. This latter machine is the most completely electrified lathe now offered in this country. By means of magnetic clutches and brakes, limit switches, tell-tale lamps, and similar devices,

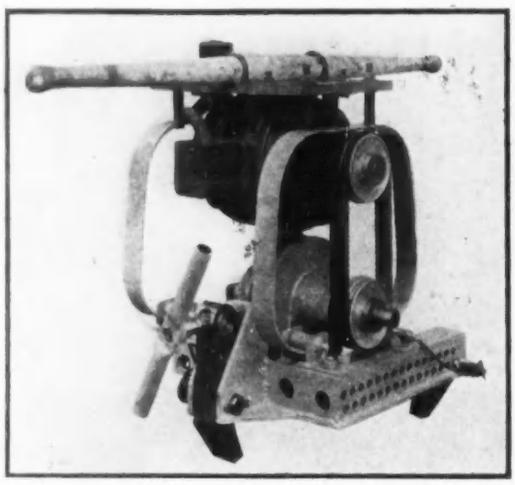
operation of the lathe is made practically full-automatic.

It is reported that at least one machine tool manufacturer will announce electrically operated and controlled machines of another type, some time in January of next year. Features of these new machines include the exceptional sensitivity of the controls, and the speed and flexibility of operation obtained through electrical and mechanical interlocks.

On the other side of the picture is the rapid strides being made in the design of new hydraulically-operated machines, especially in the machine-tool field. As a result of work in research laboratories, new control devices and better pumps have been developed. The next year will witness a fast race between the exponents of electric operation and those of hydraulic operation.



(Right) Primarily to assure a permanent lustrous black finish and light weight, the Bausch & Lomb Optical Company make extensive use of molded black bakelite in their new design of microscope



Machine to vibrate structural members to improve bonding with concrete

• *Arc-Welded Portable Electric Vibrator*

Developed to vibrate structural steel members to compact the concrete in which they are being embedded, the vibrator shown in the accompanying illustration consists of the vibrating head and a motor, both of which are mounted with spring suspension to absorb vibration. Clamps attached to the frame permit the attachment of the machine directly to the structural member without additional support of any kind.

Vibration is produced by an eccentric weight rotated at a speed of 2,600 r.p.m. around an axis parallel to the structural member to which the machine is clamped. The amount of vibration required for various sizes and weights of members vibrated, can be adjusted by means of an adjustable disk. Power is supplied by a 3-hp., 3-phase, 220-440 volts Lincoln motor.

Skewing of the machine while in operation is prevented by clamps attached to the beam. Two jaws which can be adjusted for beam widths of 15 to 18 in. hold the machine on one side. A screw clamps on the opposite side, locks it in position. A long pole handle on top of the machine is provided for moving it.

According to the manufacturer, the International Steel Tie Co., of Cleveland, Ohio, arc-welding was found to be the only method of construction which would enable this machine to withstand the severe vibrational stresses to which it is subjected and at the same time make possible a machine that would be light enough to be moved by two men. The frame must bear the terrific vibration of the machine, since it is clamped directly to the structural member being vibrated. It is not only a problem of the magnitude of the stresses, but fatigue endurance as well. According to the manufacturers, the arc-welded frame has withstood every condition to which the equipment has been subjected.

• *Recent Trends in Textile Machinery*

Although oilless bearings have been in use for a great many years in textile machinery, recent months have witnessed a definite trend toward a wider adoption of this type of bearing. Because of the extreme difficulty of washing oil spots out of rayon, the Atwood Machine Company has adopted oilless bearings for their winders, double-twisters, twisters and redraws. Most of these oilless bearings are metal, although oil-impregnated wood bearings are use for the lightest applications. In a few instances, combination wood and metal oilless bearings are used.

Another important factor that dictated the adoption of oilless bearings is the elimination of the necessity of attention to lubrication. The importance of this can be judged from the fact that a 120 spindle winder has 544 bearings, while a 200 spindle twister has 96 bearings.

Along with the general adoption of anti-friction bearings, a new development is the use of ball bearings without an inner race, the hardened shaft serving as the inner race. This design is only for small shafts carrying light loads.

Since the introduction of variable speed drives about a year ago, there has been a marked trend toward the general adoption of variable speed drives for warp beamers, pickers and slashers. There also has been introduced a variable speed drive for looms.

Previously, the low speed of many textile machines has been an obstacle in the way of the adoption of individual motor drives. This

has now been largely overcome through the use of gear motors, which have greatly extended the application of individual drives.

• *New Chest-Type Refrigerator*

Features of the new lift-lid chest-type Leonard refrigerators include an eight-position temperature selector that is located on the front of the cabinet; two freezing trays, producing 3.2 lb. of ice; a stainless porcelain bottom in the food compartment; a removable basket for the accommodation of fruit, eggs and vegetables; a white porcelain table top; a specially designed balloon door gasket that insures perfect sealing; and a patented door check that eliminates hinge strain.

The cabinet is mounted on legs $2\frac{3}{8}$ inches high that are equipped with gliders to prevent marring of floors. With the exception of the white porcelain table top, the exterior is finished in white lacquer on Bonderized steel. The interior is white porcelain.

The new refrigerator, to be known as the Leonard Electric Chest, is $32\frac{5}{8}$ inches high, $27\frac{7}{8}$ inches wide and 19 inches deep. Shelf area totals 4.3 square feet and net volume is 2.00 cubic feet. By means of an ingenious arrangement of ice trays, evaporator and shelves, the chest comfortably accommodates a quantity of food far larger than is generally expected in a refrigerator of this rated capacity. According to the Leonard Refrigerator Company, this new model is to retail at $77.50 list. This low price is indicative of the engineering ingenuity required to design mechanical equipment for the home to sell at a low price but still possess all the features of the larger and higher priced models.

By careful design, practically all of the features of larger models were incorporated in this chest-type refrigerator

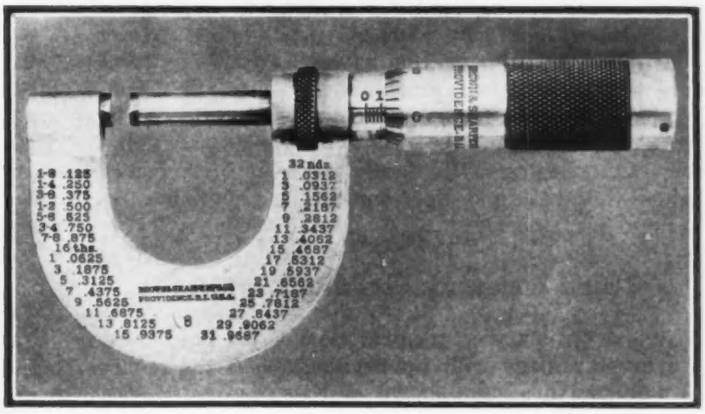
• *Hard-Facing Increases Durability*

Two-wheeled trucks, such as used in meat packing plants, are usually brought to a stop by dropping the legs on the concrete floor. But the braking action of the metal sliding on the concrete results in rapid wear on the bottom of the truck legs. About every six months it becomes necessary to take the truck out of service and attach pieces of plate to the legs. These plates are about $1\frac{1}{4}$ x 3 in., and are made of $\frac{1}{4}$ in. boiler plate bent to the exact contour of the legs.

In an attempt to overcome the rapid wear on the truck legs, such trucks have been fitted with leg plates that had been coated with Haynes Stellite, a wear-resisting cobalt-chromium-tungsten composition. After one year in service, the hard-faced wearing plates have been found but slightly worn. Measurements indicate that they will last approximately seven times longer than the ordinary wearing plate made of plain carbon steel.

Wear is also an important factor in the life of micrometer calipers. For continued accuracy in measuring over cutting surfaces, such as twist drills or grinding wheels, or in measuring across surfaces carrying imbedded abrasive, surface wear soon destroys the accuracy of micrometer calipers.

To reduce wear and increase the life, tungsten carbide measuring surfaces are used in the design of Brown & Sharpe calipers. Tungsten carbide tips, shown in the accompanying illustration, are sweated to the anvil and spindle with silver solder. After grinding to diameter, the faces are lapped square with a brass disc charged with diamond dust.



Tungsten carbide tips on both anvil and spindle greatly reduce wear and abrasion in Brown & Sharpe micrometer calipers

Actual tests made by Brown & Sharpe show that faces of tungsten carbide wear 18 times as long as conventional drill rod spindles and anvils. This increase in the service life is said to more than pay for the increased cost of attaching and finishing the hard-facing material.

Instruments and Methods

Of a Machine Tool Research Laboratory

Part 2

HANS ERNST and MARIO MARTELLOTTI
Research Department, Cincinnati Milling Machine and Cincinnati Grinders Incorporated

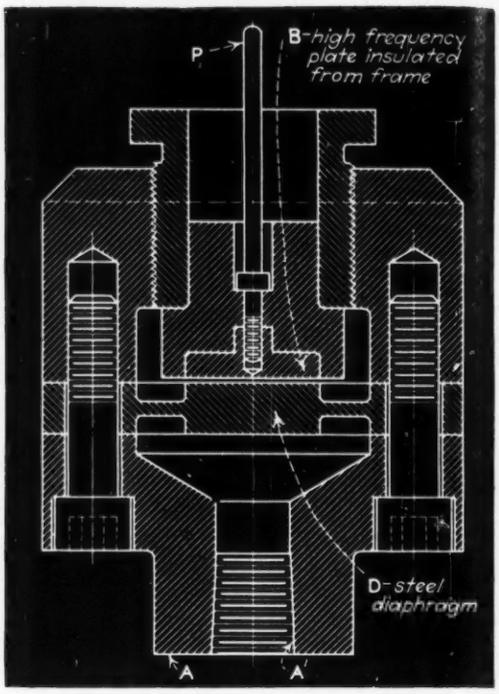


Fig. 4—Capacity-type pressure gage to record high frequency pressure fluctuations

IN THE first part of this article *(Sept. P. E.)* was described the application of the kinematograph, oscillograph and accelerometer for the detection and measurement of mechanical vibrations in machinery. In this second and concluding part is described the use of two more instruments that have been found of great value in the study of machine vibrations, the high-frequency pressure recorder and the stroboscope. The former is used for the study of rapidly changing hydraulic pressures, a matter of increasing importance because of the rapid developments in hydraulically-operated machinery.

High Frequency Pressure Recorder

Another primary instrument which has been used to great advantage with the oscillograph is the capacity type pressure gage shown in Fig. 4. This device is used in the study of rapidly changing hydraulic pressures, and with it pressure fluctuations as high as a thousand cycles per sec. can be recorded on the oscillograph.

As shown in the drawing, the body A is provided with a pipe thread A' for connection to the pressure line under investigation. A steel diaphragm D is securely fastened to the body, and so arranged that any fluctuation of pressure in the line will produce a variable deflection substantially perpendicular to the axis of the body. An insulated plate B is carried by an adjustable plug P so that it may be brought to a position parallel to the upper surface of the diaphragm D, and spaced therefrom by any desired small amount.

In operation, the plate B and the diaphragm D form a condenser which is connected in circuit with a specially developed vacuum tube oscillator, operating at a frequency of about 1,800 kilocycles. The entire circuit is so arranged that a change in capacity produced by a deflection of only 0.00001 in. will produce a measurable

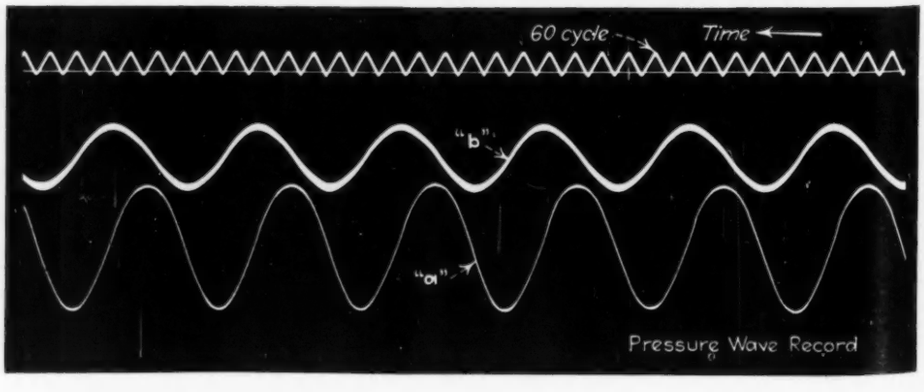


Fig. 5—Oscillograph record made with two capacity-type pressure gages, in a study of pressure wave propagation in tubes. Curve "a" shows pressure fluctuation at ingoing end, curve "b" at outgoing end

change in the current through the oscillograph element. With a properly designed circuit, the deflection of the oscillograph element is very nearly proportional to the pressure change when working within the range of pressures for which the diaphragm is designed; thus the record made on the oscillograph film will be a true reproduction of the pressure fluctuation.

A simultaneous record made with two of these pressure gages in the study of pressure wave propagation in tubes of various diameters and lengths is shown in Fig. 5. Curves (a) and (b) show respectively the pressure fluctuations at the ingoing and outgoing ends of a tube. The phase differences between the curves are a measure of the velocity of propagation through the tube, while the difference in amplitude represents the energy loss.

High-Speed Stroboscope

Another instrument which has many uses, particularly in the study of fast moving, recurring phenomena, is the stroboscope or vibroscope. This instrument is based on the well-known principle of persistance of vision which makes a rapid succession of images of an object appear as a continuous image. Fig. 6 shows an application of this instrument to the study of link and sprocket engagement in a rapidly moving chain. In this application successive images of the chain are formed by illuminating the latter by means of a Neon lamp of the type now commonly used in street signs and show window displays. In this case, however, the voltage impressed on the lamp is interrupted by a contact breaker driven by a small variable-speed motor, thus producing a very rapid succession of light flashes which appear to the eye as a continuous light. The contact breaker is so designed that the Neon tube is illuminated during only a small portion of each cycle.

When the contact breaker is driven at such a speed that we obtain one flash of the Neon lamp during the time of passage of one link of the chain, then it is obvious that the time between successive flashes will be just sufficient to permit the chain to advance a distance equal to one link, and so the chain will appear to be standing perfectly still instead of moving at high speed. By a manual adjustment of the stationary member of the contact breaker we may thus obtain an image of the chain in any portion of its cycle. By a slight change in the speed of the contact breaker, the succession of flashes may be made either slower or faster than the succession of link passages; thus the chain may be made to appear to be moving as slowly as we wish, either in the forward or backward direction.

This form of stroboscope may also be used for the measurement of vibration frequencies by merely illuminating the vibrating member from the Neon lamp and adjusting the speed of the contact breaker until the motion of the vibrating object appears to stop. The frequency of vibration may then be directly determined from the speed of the contact breaker as indicated by the tachometer.

One limitation of the stroboscope is that it cannot be used when the machine element is moving too slowly. It is necessary that the time between successive flashes be less than the duration of the persistence of the image in the eye. Commercial moving pictures run 16 frames per second. Less than about 10 flashes per second does not give enough duration of light to form a good image.

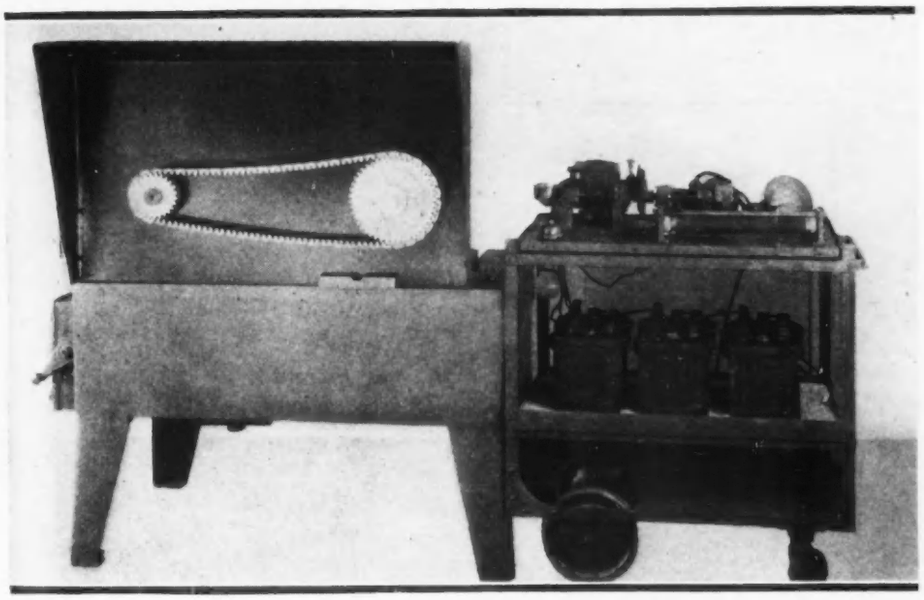


Fig. 6—Application of high-speed stroboscope to the study of the link and sprocket engagement in a rapidly moving chain. The chain can be made to appear to be standing still or moving as slowly as may be desired, in either direction

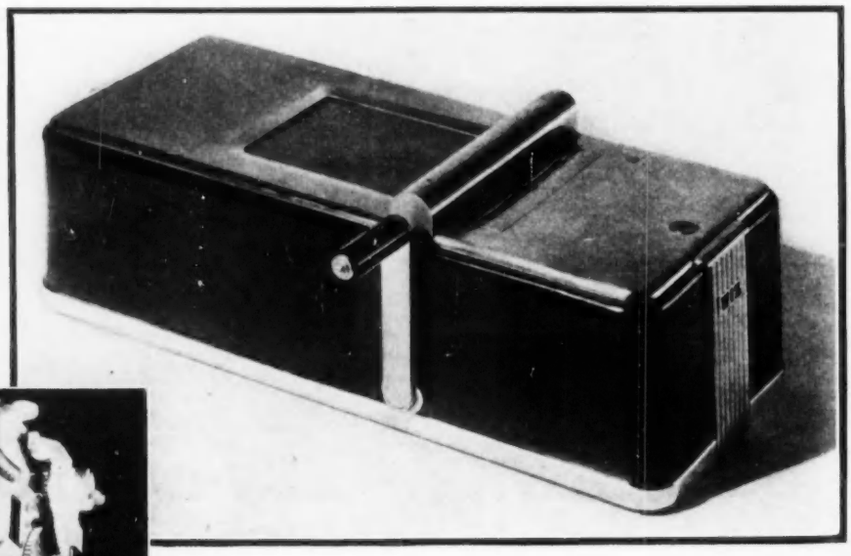
In the progress of our civilization, which has been in large measure gaged by the ability of men to transform the material riches of the world into tangible, useful things, the machine tool has played a prominent rôle. In its continued development it will further help our general advance.

The machine-tool builders who are conscious of the responsibility of their position, have introduced into their organizations the methods of investigation and analysis which once were found only in scientific laboratories. As a result such organizations stand ready to furnish the machines which will meet the new and more exacting requirements of tomorrow.

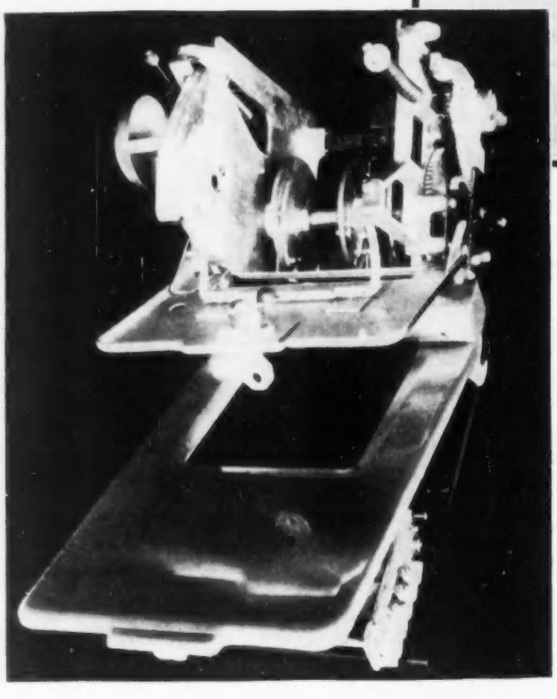
The Economic Selection of Metal Parts in Product Design

LOUIS H. MORIN
Pack-Morin, Inc.

Only by selecting the material and type of construction that will be most economical when taking advantage of all its useful properties, can the most efficient and economical design of product be developed



In the design of this sales register sheet metal was selected for the housing, with ornamental die-cast escutcheon plates that also contribute to its rigidity. Below is shown the mechanism which is mostly die-castings



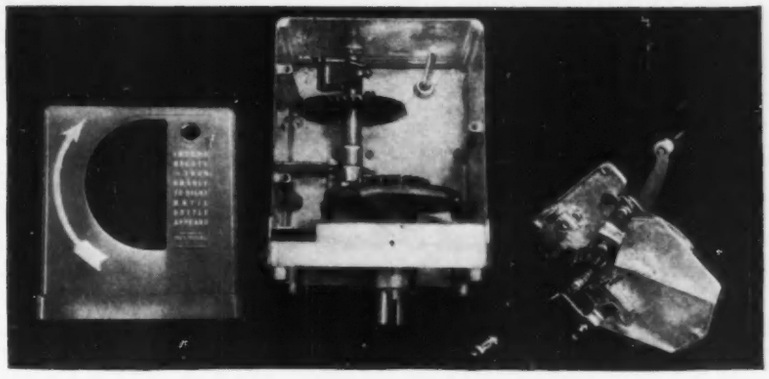
DURING the past year, many manufacturers have come to realize that their existence depended upon a better knowledge of the latest developments in the art of fabricating metals and such manufacturers have not hesitated to redesign their product where necessary. The problems of redesigning a mechanical appliance to permit its manufacture in the most economical manner is not the problem of the foundryman, the die-casting man, the stamping shop or the screw machine engineer. The problem can be solved in the best manner only by the engineer who is thoroughly familiar with the advantages and shortcomings of all methods of production and who is ready to use any of these products where they can be used to the best advantage.

No attempt will be made here to compare the proper-

ties of die-castings with those of stampings or screw machine parts or sand castings. Such information is readily available in textbooks and technical literature. Two practical examples of product redesign typical of the problems facing many manufacturers today are illustrated in the method of procedure followed in these cases. In the one, the problem was one of improving the appearance and functions of a mechanical device that had been on the market for many years, while in the second case the problem was to develop a new device at the lowest cost.

The American Sales Book Company has for many years marketed a sales register which was manufactured by their subsidiary company, the U.P.M. Kidder Press Company. Recently the executives of the American Sales Book Company decided that to stimulate the sale of their register, it was necessary to redesign it both as to appearance and mechanical functions. Walter Dorwin Teague was selected to "modernize" the sales register. The results of Mr. Teague's work are shown in the accompanying illustration.

In this vending machine, pressed steel, cast iron and die-castings are used, each where they serve the purpose most economically. The parts of the coin-operating mechanism shown to the right are mostly die-castings except the large lower gear which is cast iron



The sides of this machine are finished with a black, rubberoid enamel, the cover being finished in a satin chrome as are also the bottom strip and the front and rear escutheon plates. These plates carry the trade-mark "Wiz" in depressed engraving which is filled in with red filler. It is interesting to note that although the housing covers an intricate mechanism, no screw or bolt heads are visible on the machine. The mechanism is assembled on a chassis which is secured to the bottom of the housing. By unlocking the machine, the cover can be opened to adjust the mechanism.

The design of the machine as submitted by the artist was fine in appearance, but the commercial production of a machine to conform to the artist's ideas presented many problems not heretofore encountered. C. J. Manuel, the chief engineer of the Kidder Press Company, finally came to the conclusion that in order to carry out the artist's ideas it was necessary to scrap all the tools used on their register and redesign the mechanism completely. Although his previous experience had indicated that die-castings were relatively more expensive than stampings, he approached the problem of redesigning his register with an open mind, calling on experts in the various fields of metal fabrication for advice.

The results obtained were quite startling. Not a single sand casting was used on the new job. The entire housing was made from sheet metal stampings with the exception of the semi-circular cross member shown on the cover of the machine, which is die-cast. The two main frames of the chassis are die-cast as are also some 30 other parts.

It was found that the use of zinc base die-castings made it possible to produce a product that would meet the mechanical as well as the cost requirements. The time element was also a prominent factor and here again the die-casting process "saved the day." It is interesting to note that the tools for all die-castings were completed and samples of all parts were in the hands of the Kidder Press Company four weeks after the order was placed. Furthermore, the use of some die-castings tended to simplify the stamping problem and reduce the cost as well as the delivery time of a number of the stampings.

The housing of this register is approximately 18 in. long, 6 in. high and 7 in. wide. It is possible to die-cast this housing, but such a die-casting would be heavier, more expensive and involve a higher finishing cost than a sheet metal housing. A careful study of the factors involved indicated that this housing could best be produced as a sheet metal stamping, in spite of the difficult deep drawing operations required and the substantial waste in sheet metal involved.

The ornamental effect of the front and back plates

with the word "Wiz" could not be made integral with the sheet metal housing, but required the application of an additional ornamental escutcheon plate.

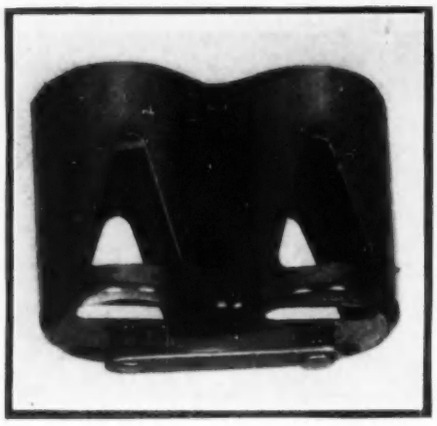
The sides of the housing were made from two simple flat sheet strips with no waste of sheet metal. The escutcheon plate was zinc die-cast, four bosses cast integral on the back of this casting. These bosses were threaded and fitted into four corresponding holes in the sheet metal stampings. The die-cast escutcheon plates besides being highly ornamental act as tie rods, making a rigid housing with the aid of the two simple and inexpensive stampings.

Another product that illustrates the importance of the proper selection of metal parts is the design of a vending machine for bottled "Coca-Cola." Vending machines for bottled beverages have been on the market for some time, but the executives of the "Coca-Cola" Company decided that the sale of their product could be increased by a vending machine exclusively their own.

Everett Worthington of Chicago, was called in to create the art design. The machine is finished in a bright, "Coca-Cola" red, the letters embossed in the sheet metal housing being finished in white.

The vending mechanism consists of an endless chain conveyor which carries the bottles through the cooling area and by proper connection with the coin operating mechanism, ejects one bottle of cold "Coca-Cola" when the proper coin is inserted. With ice refrigeration every available inch of space is utilized for either bottle or ice capacity since it is essential that the ice chamber hold a sufficient quantity of ice for 24 hours service. Where electrical refrigeration is used, the refrigerating unit is underneath the cooling chamber, the cooling coils being directly underneath the bottles.

In the design of this vending machine the cabinet walls



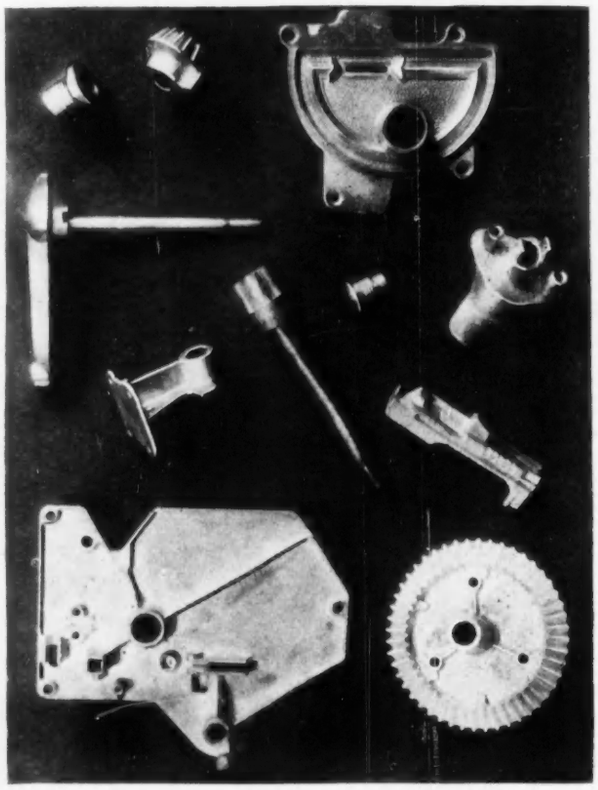
It was first considered that sheet steel cups would be too expensive because of tool charges. But by this "twin" design they are made more economical than cast iron

are logically made from sheet metal stampings, but the handles, drain cocks and other trimmings are zinc die-cast.

Referring to the coin operating assembly, practically every part is zinc die-cast with the exception of the larger lower gear which is integral with the operating shaft and is made of cast iron. The sand casting is more economical because the gear teeth as cast are sufficiently accurate for the purpose, which leaves very little machining to be done on the sand casting. The casting weighs about five pounds and the possible saving in machining costs cannot compensate for the added basic metal cost of the zinc die-casting.

In the design of the delivery mechanism, no die-castings were used since this mechanism is constantly immersed in cold water which would react against the life of a zinc base die-casting. The chassis of the delivery mechanism was made from sand castings. The determining factor in this case was tool charges. The cost for tooling up for the stamped chassis would be about ten times that of the sand cast job.

The cups were submitted to some leading stamping



Die-castings used for the coin-operating mechanism of the vending machine. Here is shown how the use of inserts makes for economy

companies, but their cost estimates indicated that the use of a sand casting would be more economical. However, the stamping companies were unwilling to concede their failure on this job and before production on the vending machine was begun, a sheet metal, stamped cup was developed which showed a substantial saving over the sand casting. To reduce costs, two cups were incorporated into one stamping. The large triangular holes on the sides are to permit free circulation of the cooling medium while between the two cups, a small rectangular tab is punched inwardly from each side. These tabs are spot-welded together tending to produce a more rigid unit. Experience with the design of this cup illustrates the importance of keeping after a job, and not assuming that one metal product is more expensive than another on the basis of a mere superficial estimate of cost.

CAST IRON
The Dean of Ferrous Metals

A survey of the facts and formulas that must be considered in developing sound designs to be made of cast iron

J. M. LESSELLS
Consulting Engineer

THERE is no doubt that the reason cast iron is widely used in engineering design is because it is easy to produce, has a low cost relative to other materials, and has a wide range of physical properties. In the swing of the pendulum several years ago to the adoption of welded construction the above facts were apparently lost sight of in some instances. Today there is plenty of evidence, however, that there is a decided approachment to the use of cast iron, especially for those applications where large quantities of the same part are required.

If a design is to be sound, it is essential that the engineer should have very clear ideas on the factors governing the choice of working stresses. These various factors are: (1) accuracy of formulas; (2) dimensions of members; (3) estimation of forces, and (4) characteristics of the materials used.

As regards the accuracy of formulas, it is our belief that in many cases formulas represent in a precise manner the actual conditions. Of course, we would recommend that more detail consideration be given to the method of stress analysis by the application of the more advanced elasticity theory.

As regards dimensions of members, this should offer no difficulty since they are easily determined. But the estimation of the magnitude of the active forces is usually difficult to determine. For instance, high stresses may occur without the action of external forces, such as residual casting stresses. Stresses may also arise from statically indeterminate systems, temperature variations or even inaccuracies in assembly. As regards the characteristics of the material, these are important since on them depends the extent to which the values obtained in a laboratory can be applied generally. It seems that it is only by a proper selection of control tests in conjunction with fundamental data that the necessary assurance can be given the designer.

From the above, it becomes apparent that, in the main, there are two types of tests. The first is tests that provide fundamental data which can be used in design. To this class belong the tensile, compression and fatigue tests. The second type is one for controlling the material product. In this latter class fall impact, torsion and modulus of rupture tests. This latter type of test is of especial importance in the case of cast iron.

Material Data from Fundamental Tests

With reference to fundamental data, excellent results have been given on the properties of cast iron by Kommers in a paper "The Static and Fatigue Properties of Some Cast Irons," *Proceedings of the A.S.T.M., 1928.* The more important of these data in so far as they refer to our discussion are reproduced here. Table I

Table I—Per Cent Chemical Composition of Cast Irons

Series	Total Carbon	Free Carbon	Combined Carbon	Silicon	Sulfur	Phosphorus	Manganese	Nickel	Chromium	Vanadium	Titanium
A	3.30	2.66	0.64	1.12	0.090	0.21	0.62	———	———	———	———
B	3.31	2.70	0.61	1.45	0.106	0.367	0.86	———	———	———	———
C	3.51	2.82	0.69	1.58	0.088	0.72	0.54	0.00	0.01	0.05	0.06
D	3.23	2.66	0.57	1.96	0.072	0.44	0.54	———	———	———	———
E	3.24	2.84	0.40	2.85	0.101	0.168	0.57	———	———	———	———
F	3.46	3.02	0.44	2.35	0.078	0.27	0.62	0.63	0.22	———	———
G	3.35	2.85	0.50	2.25	0.084	0.26	0.60	———	———	———	———
H	2.98	2.24	0.74	2.22	0.082	0.057	0.52	1.15	———	———	———
J	3.68	3.13	0.55	2.17	0.080	0.646	0.54	———	———	———	———
K	2.86	2.21	0.65	2.51	0.079	0.057	0.62	———	———	———	———

Table II—Summary of Tensile and Fatigue Tests

Series	Tensile Strength lbs./sq.in.		Compression Strength lb./sq.in.	Brinell Number	Endurance in Fatigue lb./sq.in.
	Small	Large			
A	35,700	35,100	109,000	202	19,000
B	39,500	39,700	137,600	207	19,000
C	30,300	32,300	117,600	193	15,000
D	31,900	30,400	122,800	192	12,000
E	31,900	31,100	117,000	184	13,400
F	37,000	35,600	132,000	209	19,500
G	35,600	33,700	130,900	197	19,100
H	51,000	48,100	156,300	238	22,000
J	23,200	24,900	85,000	171	11,800
K	42,500	50,300	154,700	237	24,100

gives the chemical composition and Table II the corresponding physical values.

The tensile specimens were of the standard types having enlarged threaded ends, the smaller specimens having a diameter of 0.330 in. at the center and the larger a diameter of 0.600 in. It will be noted there is remarkable agreement in both sets of tests. The series *H* and *K* are high-strength cast irons. Series *H*, as will be observed from Table I, contained 1.15 per cent nickel. The endurance limits given in Table II were obtained from beam specimens rotated in a fatigue testing machine at 1,450 r.p.m. and represent values for complete stress reversals. It should be observed that the endurance limits for these two irons equals that of low carbon steel.

Kommers states, which is of importance to designers, that there is evidently more variation in the tensile and fatigue results than in the transverse and compression tests, because such tests may be more sensitive in picking out flaws and weaknesses, should such exist.

Impact, Torsion and Modulus of Rupture

For the second type of tests, namely, control tests, the recently published "Symposium on Cast Iron," *Proceedings of the A.S.T.M., 1933*, p. 117, gives an excellent idea of what is being done in this direction. Although this symposium is directed more towards such tests, it contains excellent information useful to the designer dealing with the general physical properties of cast iron. Referring specifically to those mechanical properties being discussed here, it is now possible for the A.S.T.M. to classify, at least tentatively, cast irons on the basis of minimum tensile strengths, Class 20 being 20,000 lb. per sq.in., Class 25 being 25,000 lb. per sq.in., and so on in seven stages to Class 60 with a tensile strength of 60,000 lb. per sq.in.

Some other data of useful design nature shows the relation between tensile and compression strength. From this data it can be concluded that as the tensile strength of cast iron is increased there is a decrease in the ratio of compressive strength to tensile strength. For an iron having a tensile strength of 15,000 to 20,000 lb. per sq.in. the ratio of compressive to tensile strength is given as 4.08, whereas for an iron of tensile strength 55,000 to 60,000 lb. per sq.in. this drops to 2.45.

Some data on the subject of creep are also given from which the authors conclude that the higher strength cast irons may be useful considerably above the temperatures which engineers are accustomed to consider for cast iron. The above conclusions are advanced on the grounds that a plain cast iron of 45,000 lb. per sq.in. tensile strength showed a creep of only 0.0025 in. per in. during 1,000 hr. at a stress of 10,000 lb. per sq.in. at 600 deg. F.

From all the above considerations—accuracy of formulas, accuracy of dimensions, estimation of forces and material characteristics—it will be apparent that the working stress adopted must be much less than the stress at which failure of the material is to be expected. In selecting working stresses a factor of safety is, therefore, employed which is defined as the factor by which the working stress must be multiplied to give the stress to produce failure. In recent years a rational approach has been made to the choice of working stresses for ductile materials such as steel. There is no reason why the same approach cannot be made for such materials as cast iron, particularly so since it has been shown that it is now possible to get irons of a tensile and fatigue strength comparable with the lower carbon steels.

If we are considering the application of cast iron to simple static stress then the ultimate strength in tension rather than the yield point, as for ductile materials, can become the basis of choice, and if the application is one of stress variation, then the endurance limit can be used. For those cases of combined stresses, either static or dynamic, the various strength theories may be used for cast iron as they are now used for the so-called ductile materials. This question is discussed more fully in "Applied Elasticity," Timoshenko and Lessells, *Westinghouse Night School Press, 1925.*

Cast iron is always classed as a brittle material, but then we are not sure where a ductile material ends and a brittle one begins, as shown for instance by the tensile test. In each and every case the working stress can only be a fraction of what the material can bear and it is to be expected that the precaution taken by the designer in the choice of safety factors will be greater for cast iron than for steel. The rational approach to the choice of working stresses such as has been discussed here is what is recommended.

Recent developments in the improvement of parts, materials and finishes, and how product engineers are using them, will be told in the November issue

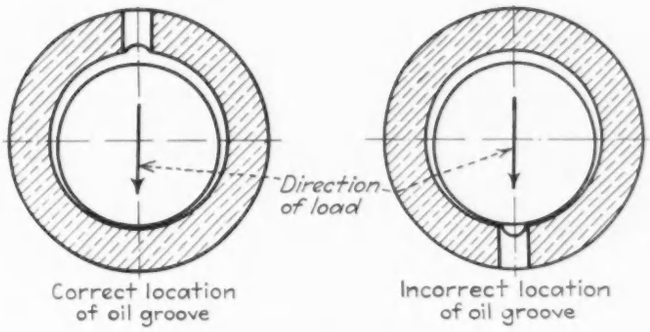


Fig. 1—Oil grooves should be located on the side of the bearing where the pressure is low

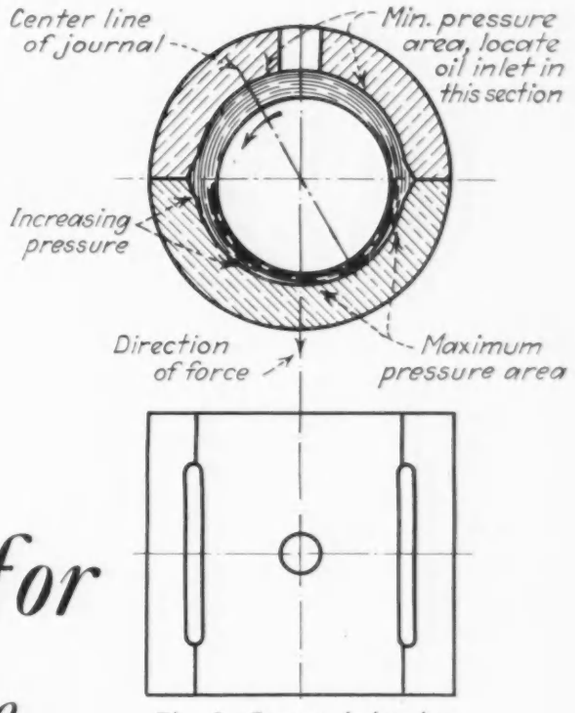

Center line of journal
Min. pressure area, locate oil inlet in this section
Increasing pressure
Direction of force
Maximum pressure area

Fig. 2—Correct design for a well-lubricated bearing, showing the proper location of oil grooves and oil holes

Possible Causes for Bearing Failure

C. H. LEIS
Chief Engineer, Johnson Bronze Company

In this last article of the series on sleeve-type bearings, the author explains the design of oil grooves and points out possible causes for bearing failure

IF WE list ways in which a bearing may fail to give satisfactory service, we may examine the methods of combating them and see to what extent engineering design and construction are concerned, and how far metallurgical testing may afford control of the variables. One might also examine the testing problem to see whether test methods are available to measure the really important properties and how far specifications, as now drawn, may insure getting acceptable bearings.

When and How to Use Oil Grooves

One common cause of bearing failure is the improper design and location of the oil grooves. Simplicity should be the keynote in grooving for all types of bearings. Oil grooves should not lead into the load-carrying area. That will decrease effective bearing area, destroy oil film continuity, conduct lubricant away from where needed most and destroy the film pressure.

It is also proved that lubricant cannot be taken in by the bearing if the oil-hole is within the load-carrying area. Therefore, as shown in Fig. 1, the oil grooves should be located where the pressure is low.

In considering whether or not to put oil grooves in the bearing, the common operating factors of speed, load, bearing length and type of lubrication should be taken into account. At high speeds and light loads, or with pressure oil feed systems, grooving is not necessary, but is needed with high loads or slow speeds.

In comparatively long bearings, grooving is of importance to distribute the lubricant the entire length of the bearing. In all cases, longitudinal grooves, as in Fig. 2, are most desirable, but they should not be cut through to the ends of the bearings, except where the lubricant is introduced at the ends. When the loaded side of the bearing cannot be predetermined or is not fixed, a circular groove is preferable.

Oil grooves in thrust washers must be designed so as to establish a wedge, pointing towards the direction of rotation. This is illustrated in Fig. 3. A common but incorrect type of groove sometimes used in thrust bearings is shown in the same illustration. When lubrication is applied from the outside, the grooves may be cut to the outer edge of the washer. In no case should the washer have a circular groove. Such a groove would act merely as a circulating duct and tend to prevent building an oil film pressure.

Causes for Excessive Friction

A bearing may show high starting torque, because of wrong clearance or poor finish, or because of having lost its oil film while under the static pressure of the stationary shaft. Obviously the better the fit and finish,

the lower the static pressure per unit area and the less the tendency to squeeze out the film. The ability of the bearing (or shaft) to absorb and hold an oil film may be a specific property of the alloy with respect to a given pure or compounded oil, a property more or less allied to chemical affinity.

Actual porosity for oil retention may be obtained at the sacrifice of some other properties. Some idea of porosity of a cast alloy can be had by density determinations before and after compression.

Chemical Reactions

Failure might occur as a result of chemical reaction between the bearing or shaft and the lubricant, resulting in attack on the metal or deterioration of the oil. Catalytic action on the oil might occur because of its contact with metal.

Failure by Friction

A bearing may show high running friction. This may be because of: (1) poor finish, projections on the bearing cutting through the oil film; (2) insufficient clearance at operating temperature, which refers back to the thermal expansion of the journal bearing and its support and their relative and absolute dimensions; or (3) the use of a lubricant of wrong viscosity.

Aside from thermal conductivity, this problem is relatively independent of the bearing material. When thermal conductivity is a factor, it can be measured. The bearing dimensions are a matter of engineering. Lack of adequate lubrication under running conditions is entirely a matter of design and not of the bearing material. How the bearing metal acts when lubrication does fail is, however, a function of the material.

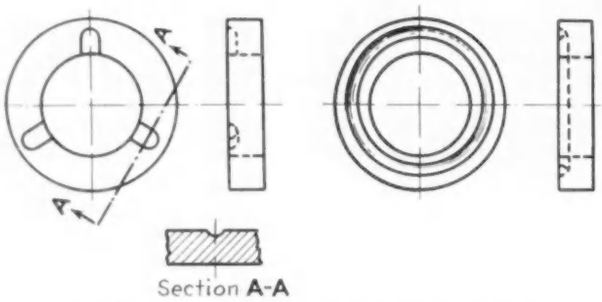


Fig. 3—In thrust bearings the grooves must run radially, not concentrically. The latter would decrease the effective bearing area without contributing to oil distribution

Scoring or Seizure

Scoring of the shaft, or actual seizure, will again depend partly on the relative thermal expansion of the materials in the bearing assembly, and partly on whether or not the contacting materials will weld together. Dissimilar materials are usually chosen for shaft and bearing. Sometimes specially treated intervening surfaces, such as nitrided or case-hardened surfaces, may be necessary to prevent seizure. It is usually desired to have the shaft wear less than the bearing, hence a harder material is chosen for the shaft than for the bearing, but too wide a difference should not exist. Thus, one uses the harder bearings against the harder shafts.

If it is desired to have a lining that will melt out when the temperature rises through lack of lubrication and to have a back that will not score the shaft after the lining has melted, as in railroad practice, the melting point and plastic range of the lining are important and must be determined.

Seizure might possibly result from dimensional changes of a bearing alloy because of fatigue, lead sweats and similar factors. These can be determined by special metallurgical tests.

Plastic Flow or Conformability

Some degree of plastic flow or "conformability" may be desirable to allow the bearing to be "run in" rapidly to a good fit. Hardness, compression, and pounding tests again help here. A study of the rate of work-hardening might be useful to determine whether the deformation will continue or will promptly be brought to a stop. A low proportional limit will tend toward conformability.

Permanent Distortion

The permanent distortion and tendency to flow is a function of the elastic and plastic properties of the bearing material. The shaft bends minutely under load. The bearing should yield elastically to cushion itself to fit without undue plastic flow. The elastic deformation is a function of the relative moduli of elasticity of the shaft, the bearing and the support, and their dimensions.

The bearing may wear with actual removal of material by reason of poor fit or finish and high local unit pressure. Excessive wear may result from having metallic contact through loss of oil film while at rest, or from abrasive particles embedded in the bearing lapping the shaft.

Failure by Wiping

Tied up with wear and perhaps allied to seizure is failure by wiping. Generally wiping results from exceeding the temperature at which the alloy gets plastic. Possibly a combination of pressure and temperature causes a certain amount of sticking, approaching actual seizure. This wiping is caused by the bearing collapsing at the parting line, thereby developing a high friction at this point. Wiping, as such, hardly occurs in the Cu-Sn-PB system of alloys, though in highly leaded alloys the lead may sweat out as a result of high temperature.

Failure by Fracture or Cracking

The bearings may fail by fracture or cracking. Ductility in the tension test, the appearance of the sample in the compression and pounding tests, or still better, the impact test, tell something about this. Fatigue failures might occur in mountings where flexure is possible, or might even occur within the elastic range from repeated purely elastic deformation, or from differential temperature effects.

The Significance of WR^2 And How To Calculate It

JOHN W. HARPER
Industrial Engineering Department General Electric Company

For frequent starting and stopping, or reversal of rotation, the size of motor may be dictated by the power needed for acceleration, as determined by three factors, (1) the moments of inertia of the moving parts referred to the motor shaft, (2) the speed that must be attained, and (3) the time allowed for acceleration

WHENEVER an electric motor is to be used in an application where it must start or reverse many times per minute, one of the first questions asked is: "What is the external inertia, or WR^2, referred to the motor shaft?" Usually this request is met with a blank look. While it is strictly a mechanical function, its importance and usefulness seem to be little known, or if known, are disregarded. Many mechanical designers seem to have trouble in calculating this quantity, and also seem to have confused ideas concerning the power required for acceleration. In reality the problem is comparatively simple, being one of kinetic energy as far as the mechanical designer is concerned.

Every moving body has stored in it an amount of kinetic energy proportional to the product of its mass and the square of its velocity. To increase the velocity of a body there must be delivered to it an amount of energy equal to the difference in the energies possessed by the body at the final and initial velocities, neglecting losses. It is evident that when large masses in a machine must be accelerated to a high velocity in a short space of time, the power required may be many times greater than that needed to keep the machine in motion after the velocity has been attained. It is also evident that changing the mass of the body has much less effect on the energy required than changing the velocity.

Considering a body of mass M moving linearly at a velocity of V ft. per sec., the kinetic energy of this body expressed in foot pounds will be:

$$E = \tfrac{1}{2} MV^2 \qquad (1)$$

In this equation M represents the mass of the body, the unit of mass being called a "slug." Numerically, M in "slugs" is equal to the weight in pounds divided by 32.2 ft. per sec. per sec., the acceleration due to gravity.

With reference to rotating bodies, the mass of the body as actually distributed around the center of rotation is mathematically equivalent, insofar as its kinetic energy is concerned, to the whole mass concentrated at a certain radius R, called the radius of gyration, from the axis of rotation. The linear velocity of the concentrated mass at radius R ft. will then be $2\pi RN$ ft. per min., where N represents r.p.m. Dividing this by 60 to reduce to ft. per sec. and using $W/32.2$ for mass M, equation (1) then becomes:

$$E = \frac{WR^2N^2}{5872.86} \text{ ft. lb.} \qquad (2)$$

This is the first appearance of WR^2. It is merely the weight of the body times the radius of gyration squared. Since the kinetic or "stored" energy in the body is directly proportional to it, it is commonly known as "moment of inertia," although strictly speaking, moment of inertia is $WR^2/32.2$. However, in developing the most commonly used formulæ, the factor 32.2 disappears or is absorbed in the constants, so that WR^2 itself remains as a convenient medium of calculation. Later we will consider ways of calculating it, but first we want to consider the torque required for acceleration. The energy as calculated from Equation (2) merely represents *the input to the rotating body itself*, and bears no definite relation to the input to the driving motor during acceleration. It is, therefore, less confusing to determine the actual torque at the motor shaft necessary to accelerate. The input from the line will then correspond to this value of torque.

This torque can be readily determined by considering the angular velocity of the body. From the experimentally established physical law, $F = MA$, which states that the force necessary to accelerate a body is

equal to the product of its mass and its acceleration, the following fundamental equation has been derived for T, the acceleration torque:

$$T = I\,\theta \text{ lb. ft.}$$

In this equation, I is the moment of inertia and is equal to $WR^2/32.2$, θ is the angular acceleration of the body expressed in radians per sec. per sec. and, assuming constant torque, is equal to the final angular velocity w divided by the time t required to accelerate from rest. Thus where N is the final anguar velocity in r.p.m.:

$$\theta = \frac{w}{t} \text{ or } \theta = \frac{2\pi N}{60t}$$

Then, substituting in the fundamental equation:

$$T = I\,\theta = \frac{WR^2}{32.2} \times \frac{2\pi N}{60t}$$

or

$$T = \frac{WR^2 N}{307.6t} \text{ lb. ft.} \qquad (3)$$

It should be noted here that Equation (3) assumes constant angular acceleration up to the final velocity of the machine. Therefore, if the acceleration is not constant but increases or decreases as the machine gathers speed, this equation will not give accurate results. Nevertheless, even though the acceleration does vary while the machine is coming up to speed, it is possible in some cases to make use of this equation, as will be explained later.

It should also be noted that Equation (3), as stated above, covers the condition of starting from rest. If the body had already been running at some speed N_1 and the torque necessary to increase the speed to N_2 were being calculated, the value of N in the equation would become $(N_2 - N_1)$, assuming both speeds to be in the same direction. If the speeds had been in opposite directions, it would be necessary to use the algebraic differences, or the arithmetical sum, in calculating T. However, rather than do this it is preferable to calculate in two stages, first from the original speed down to zero, and second from zero up to the final speed. The reason for this is that during deceleration the friction troque, and frequently the load torque, assist in the retardation; whereas, during acceleration these torques oppose the motor torque. Hence, the total torque required in accelerating is greater than the calculated T; whereas in decelerating it is less. Calculating in two steps, therefore, simplifies the computation and minimizes the possibility of errors in adding or deducting the friction and load torques.

At any rate, a glance at Equation (3) immediately discloses that the torque necessary to accelerate a rotating body depends directly on the value of the term WR^2—hence, on the body's moment of inertia. Thus, once we have found the value of WR^2, it is relatively simple to determine the horsepower required for acceleration, by making use of the familiar equation:

$$HP. = \frac{\text{Torque} \times N}{5250} \qquad (4)$$

In the above equations the factors WR^2 and N pertain to the rotating body or machine element. As a rule the elements run at speeds different from that of the motor. In order, therefore, to calculate the torque for a system where different parts run at different speeds, it is necessary to reduce all the components to a common base. From Equation (2) and from the law of conservation of energy, it is apparent that, neglecting losses, the energy of any part reduced to the motor speed must equal that of the part when running at a different speed. Hence, omitting common constants:

$$\text{Equivalent } WR^2 \times N^2_m \text{ (at motor speed)} = WR^2 N^2 \text{ (at some other speed)},$$

or

$$\text{Equivalent } WR^2 = WR^2 \left(\frac{N}{N_m}\right)^2 \qquad (5)$$

In other words, the equivalent WR^2 of a part running at a speed different from that of the motor is the same as the WR^2 of that part times the square of the ratio of its speed to that of the motor. It will be readily seen from Equation (5) that with a reasonably large gear ratio between the motor and the driven elements the effect of the latter on the accelerating torque is minimized—assuming of course that the driven parts run at a speed slower than that of the motor. If their speeds are higher, the accelerating torque is obviously increased.

The above discussion has dealt with rotating parts exclusively. In many applications there is a combination of rotating parts and linearly moving parts. Therefore, to determine the accelerating torque for such combinations, it is necessary to reduce the mathematical consideration of the linearly moving parts to a basis equivalent to that used for the rotating parts—consideration of the latter generally being based on the speed at the motor shaft.

The linear speed of a concentrated mass at radius R ft. will be, as stated previously, $2\pi RN$ ft. per min., where N represents the r.p.m. Then the energy is $\frac{1}{2}M2\pi RN^2$ for the rotating body and $\frac{1}{2}MV^2$ for the linearly moving body. Equating these two expressions and substituting $W/32.2$ for M to obtain pounds, we then have, again omitting common constants:

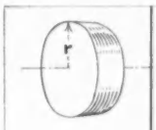
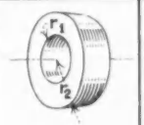
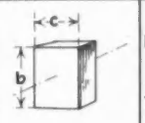
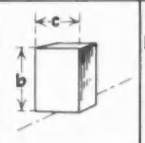
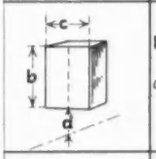
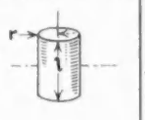
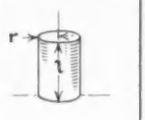
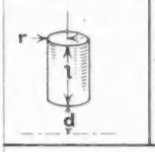
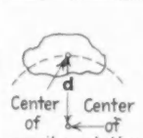
$$WR^2\,2\pi N^2 = WV^2$$

$$\text{or transposing: } WR^2 \text{ equiv.} = W\left(\frac{V}{2\pi N}\right)^2 \qquad (6)$$

In this case, the expression WR^2 represents the inertia of the linearly moving parts reduced to a basis equivalent to that of the rotating parts. It is obtained by multiplying the weight, in pounds, of the part by the square of the ratio between the velocity in feet per minute of the part and 2π times the speed of the motor in revolutions per minute.

Attention should be called, however, to the fact that Equation (6) is valid only when the velocity V bears a constant relation to the motor speed N, as in the case of a rack engaged with a pinion on the motor shaft. For reciprocating parts, conversely, Equation (6) cannot be applied directly because such parts have a varying velocity that attains a maximum in one direction, retards to zero, and then attains a maximum again in

Table I—Radii of Gyration for Rotating Bodies

	**SOLID CYLINDER** about its own axis	$R^2 = \frac{r^2}{2}$
	**HOLLOW CYLINDER** about its own axis	$R^2 = \frac{r_1^2 + r_2^2}{2}$
	**RECTANGULAR PRISM** about axis through center	$R^2 = \frac{b^2 + c^2}{12}$
	**RECTANGULAR PRISM** about axis at one end	$R^2 = \frac{4b^2 + c^2}{12}$
	**RECTANGULAR PRISM** about outside axis	$R^2 = \frac{4b^2 + c^2 + 12bd + 12d^2}{12}$
	**CYLINDER** about axis through center	$R^2 = \frac{l + 3r^2}{12}$
	**CYLINDER** about axis at one end	$R^2 = \frac{4l^2 + 3r^2}{12}$
	**CYLINDER** about outside axis	$R^2 = \frac{4l^2 + 3r^2 + 12dl + 12d^2}{12}$
 Center of gravity — Center of rotation	Any body about axis outside its center of gravity	$R_1^2 = R_0^2 + d^2$ where R_0 = radius of gyration about axis through center of gravity; R_1 = radius of gyration about any other parallel axis; d = distance between center of gravity and axis of rotation

the opposite direction—all, while the motor is running at constant speed. In such cases, obviously, much more complex mathematical considerations are necessary in order to apply the principle of Equation (6), but these are beyond the scope of this discussion.

Nevertheless, by the use of Equations (5) and (6), the calculations on all parts of a machine, that either rotate or move linearly, can be reduced to the mathematical basis corresponding to the speed of the driving motor or to any other reference speed. After the summation of the equivalent WR^2 of all parts has been made, the torque required to accelerate can be calculated from Equation (3), and the horsepower from Equation (4). The factor N in (3) and (4) is, of course, the speed of the motor, or the reference speed, since all parts have been reduced to that speed.

It will be seen from the previous discussion that WR^2 has a habit of popping up almost anywhere and that it is not only convenient but is an almost indispensable tool in the technical kit of those interested in determining accelerating torques. In fact, before any of the foregoing equations can be solved, the value of this quantity must be determined. The calculation of the weight W is simple. The determination of R, however, is relatively complicated. To simplify such calculations, Table I (left) includes the radii of gyration for various forms of bodies and for various axes of rotation of these bodies, as shown by the illustrations.

In most practical problems the actual rotating body will seldom correspond exactly to any one of the forms shown here. In every instance, however, it may be resolved into parts, each of which agrees absolutely or very closely with one of the forms in Table I. For example, a spoked flywheel can usually be separated into a rim, corresponding to Form 2; spokes, each corresponding to Form 5 or 8; and a hub corresponding to Form 2. If the flywheel is webbed, the web would also correspond with Form 2. The moment of inertia $WR^2/32.2$ of the wheel will be equal to the direct sum of the moments of inertia of all the parts as calculated separately, each about the same axis of rotation, usually the center of the shaft. Each separate calculation will simply be the weight of the particular part being considered multiplied by its radius of gyration squared. In ordinary calculations the radius is always taken in feet and the weight taken in pounds, unless otherwise specified.

In many instances it is not necessary to calculate each individual part exactly. If the flywheel has a heavy rim and is relatively large in diameter, it is usually sufficient to calculate the rim and then add a percentage for the spokes and hub, since they are lighter and are nearer the center and hence have a lower R. In the case of motor armatures, calculations without actual design data are usually based only on the weight and outside diameter. Table II on p. 398 gives values for R which are suitable for approximate calculations.

In arriving at approximate figures for the moments of inertia, the manner in which short methods involving coefficients and ratios should be used should depend on the later use which is to be made of the values so determined. In a mill table, for example, the energy stored in the rollers and the other rotating parts serves no useful purpose, but is a necessary evil incident to the use of heavy moving masses. For such an application, it is always conservative to over-estimate rather than under-estimate the moments of inertia. On the other hand, where a flywheel is used for reducing the peaks, in which case the moment of inertia of every part is useful, conservatism requires that in the consid-

(Continued on page 398)

66 Causes of Defects in Finished Steel Parts

● Steel parts are sometimes found to be unfit for use or assembly because of warpage or distortion; pitting, poor surface or scale; cracks, breaks, checks; non-uniform hardness, general softness or soft spots. It then becomes necessary to determine the cause for such a condition. The fault may lie with one or more causes, such as improper fabrication, defective material, poor design, or similar factors. To the experienced, the majority of the causes are self-evident, but sometimes it is difficult to determine the one or more reasons for a failure.

● In the following tabulation are listed 66 possible causes that might result in a steel part being unfit for service. For each cause is given one or more of nine methods of examination by which the cause of the defect can be determined.

WERNER H. C. BERG
Whitney Manufacturing Co.

STANLEY P. ROCKWELL
The Stanley P. Rockwell Co.

POSSIBLE DEFECTS IN STEEL PARTS AND HOW TO EXAMINE FOR THE CAUSE

Method of Examination

C—Chemical Analysis
E—Deep Etching
M—Microscopic
S—Spark Test
H—Hardness
X—X-Ray
F—Magnetic Testing
P—Sulphur Printing
V—Visual Inspection

Method of Examination	Defect—Pitting, Poor Surface, Scale
	Caused by Design and Fabrication
	Source of Defect:
V	Rough machined surface
V	Small metal particles adhering to surface
V-P-C	Chemical compounds from cutting, grinding, cleaning, or rust preventive fluids
V	Rust spots
	Caused by Steel
	Source of Defect:
M-P-C-X . . .	Impurities such as manganese sulphides, slag and other substances
M-C-S-X . . .	Machined surfaces of low-carbon steel scale more on heating for hardening than do high-carbon steels. Chrome alloy steels retain particularly good machined surface after heating
M-E-C-X . . .	Carburized and rehardened steels poorer than high-carbon steels
V-C.	Cobalt high-speed steel particularly poor
	Caused by Heat-Treating Operations
	Source of Defect:
V-H	Pack hardened work usually better than open-fire hardening
V-M	Heated too hot
V-M	Wet carburizing material
V-M-H	Leaky carburizing or pack hardening containers

Method of Examination	Defect—Warpage or Distortion
	Caused by Design and Fabrication
	Source of Defect:
V	Unsymmetrical design
V	Too heavy machine cuts
V	Large difference in size of cross-sections
V	Unequal removal of external surfaces
	Caused by Steel
	Source of Defect:
M-C-S	Water hardening steels more prone to warp than oil hardening steels. High manganese non-deforming types re-

	duce warpage danger. Air quenching steels have still less tendency to warp
	Caused by Heat-Treating Operations
	Source of Defect:
V	Quenching in improper position as regards symmetry
V-M-H	Pocketing air in quenching liquid
V-M-H	Too fast heating, especially of non-symmetrical parts and parts having large variations in cross-section, thereby causing red-hot thin plastic sections to be stretched by heavy sections that are still cold
V	Uneven support during heating and removal from furnace to quench
V	Uneven support during tempering
V	Uneven heat during tempering
M	Lack of proper normalizing and annealing before or during fabrication
	Caused by Finishing Operation
	Source of Defect:
V-H	Grinding cut too heavy, causing uneven expansion
M-C-S-H . . .	Unequal grinding or machining of external surfaces

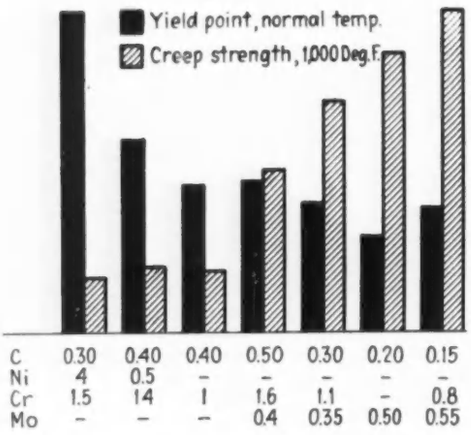
Defect—Cracks, Breaks and Checks

	Caused by Design and Fabrication
	Source of Defect:
V	Sharp fillets
V	Deep machine marks
V	Holes too near edge and too near each other
V	Proportionately large difference in cross-section
V-C-S	Wrong heat-treatment
	Caused by Steel
	Source of Defect:
V-M-F	Seamy stock
M-F-E-P . . .	Impurities such as manganese sulphides, slag, etc.
M-F-E	Internal crack, cavities, abnormal ingotism, dendrites, flakes
M	Abnormality
C-S	Wrong type of steel
M-F-E-P-S-H .	Improper forging, normalizing and annealing prior to heat-treating
	Caused by Heat-Treating Operations
	Source of Defect:
M-H	Under-heating—quenching during decalescence transformation range
M-H	Removing from quenching bath during recalescence transformation range
M	Large grain growth from over-heating
V-H	Uneven heating—Magnifying design strains caused by abrupt changes in cross-section
M-C-S	Wrong quench for type of steel—such as using water quenching for oil-hardening steel
V-H	Lack of preparation in claying, filling holes and otherwise protecting against large changes in cross-section
V-H	Unequal temperatures applied to materials prior to or during tempering
	Caused by Finishing Operation
	Source of Defect:
V-M-F	Grinding cracks
V-M-F	Pickling or plating causing hydrogen embrittlement

Defect—Non-uniform Hardness or Soft Spots

	Caused by Design and Fabrication
	Source of Defect:
V-M-C-S-H . .	Not allowing for removal of sufficient surface metal, which may be decarburized. Underlying metal will be hard if hardening has been conducted properly
V-M-C	Coppered surfaces affecting work designed for carburizing liquor finish on drawn products
	Caused by Steel
	Source of Defect:
V-M-C-S-H . .	Decarburized surface (see Design and Fabrication)
M-C-X	Carbon or hardening elements lower than supposed—general softness
M	Abnormal steel—general softness or soft spots
	Caused by Heat-Treating Operations
	Source of Defect:
M-H	Too low heat
M-H	Wrong quenching medium
M-H	Decarburizing surface atmosphere
M-H	Using carburizing material at too low temperature for reheating work
M	Worn out carburizing material
M-H	Delayed quench
M-H	Spots caused by tongs cooling work before quench
M-X-H	Retained austenite
M-H	Quenching liquid splashing on work before quench
M-H	Use of tap water containing air
M-H	Steam pockets caused by holes, tongs, insufficient motion in quench
M-H	Quenching medium too hot
M-H	Drawing temperature too high
	Caused by Finishing Operations
	Source of Defect:
V-H	Excessively heavy grinding, causing localized overheated areas equivalent to too high a draw.
V-M-C-S-H . .	Too much case removed in machining or grinding operations

Selecting Alloy Steels For High-Temperature Service



Room temperature properties of alloy steels are not a guide to creep strength at elevated temperatures

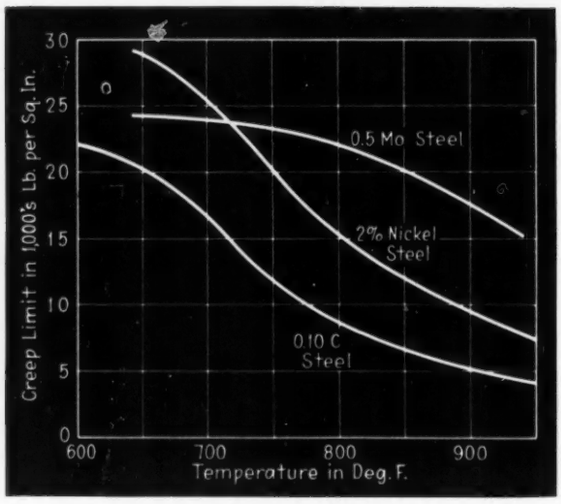
IN THE design of machine parts or structures that must withstand high temperatures, the type of alloy steel selected will be governed by the magnitude of the stresses involved and the degree of corrosion resistance required. Properties at normal temperatures are not usually indicative of high-temperature properties, as illustrated by the chart on this page, for which reason alloy steels must be selected on the basis of their performance under the heat and corrosion existing in operation.

In many applications the stresses created by the high temperature may be much higher than the calculated load stresses. Unequal thermal expansion of rigidly connected members, unequal expansion in different parts of the same member, temperature gradients, and the distribution of the metal in the part will greatly affect the actual stresses, regardless of the alloy selected.

But in resistance to corrosion alloy steels differ widely, and corrosive action is greatly accelerated by higher temperatures. Oxidizing atmospheres, high velocity steam, reducing flames, hot sulphur compounds, hydrogen and other gases each produce different types of attack.

In non-loaded structures subjected to high temperature, the principal requirement is resistance to scaling, oxidation or attack from the gases or liquids in contact with the steel. In electric furnace parts, the atmosphere usually is reducing or neutral, and for temperatures up to 1,000 deg. F., or even higher, ordinary carbon steel is entirely satisfactory. At temperatures as high as 1,600 deg. F. the addition of silicon, copper, aluminum or chromium gives an alloy steel with satisfactory life. Higher temperatures require chromium additions up to 18 per cent, but the strength of the straight chromium steel falls off rapidly so that unsupported parts may warp under their own weight. A nickel content of at least 8 per cent is therefore added to produce an austenitic iron that has greater strength for temperatures up to 2,000 deg. F., the higher temperatures requiring as much as 20 to 40 per cent nickel. In these nickel-chrome alloys the addition of up to 1 per cent molybdenum increases resistance to deformation.

In gas-fired or oil-fired furnaces, the atmosphere is often oxidizing and contains sulphur products that rapidly attack carbon steel. The oxidizing atmosphere removes carbon from ordinary steel causing intergranular attack, while sulphur compounds cause pitting of steel surfaces. Chromium additions lessen both forms of attack, but higher alloy content is necessary than



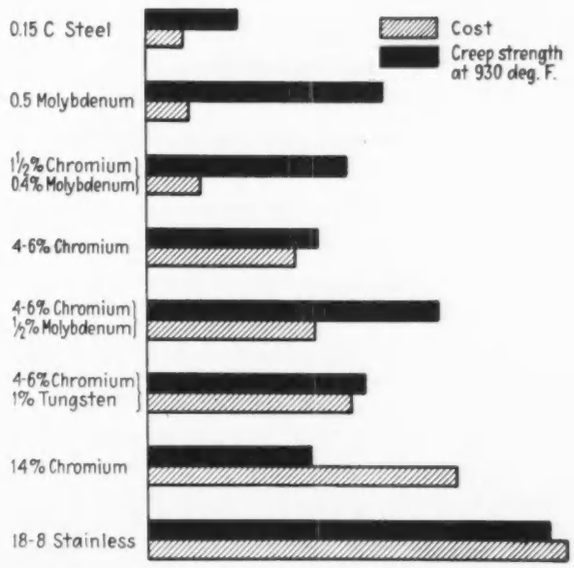
Although nickel increases the strength at normal temperatures, molybdenum has a greater effect at temperatures of 650 deg. F. and higher

required in a reducing atmosphere. Nickel impairs the corrosion resistance of steels subjected to sulphur compounds, and although some nickel may be essential for strength, the content is kept to less than half the amount of chromium.

Stainless Steel for Higher Temperatures

In atmospheres of hydrogen at high temperature, ordinary carbon steel is rapidly embrittled, producing cracks and fissures, and the rate of attack increases with the temperature. Alloy steels of the 18-8 stainless type are resistant to attack at temperatures up to 1,100 deg. F., but at higher temperatures, higher alloy content is necessary. Likewise, for resistance to hydrogen-nitrogen (ammonia) mixtures, chromium additions are effective in reducing attack at the surface. In combination with chromium, other metals such as tungsten and vanadium also add a desirable amount of resistance to this mixture of hot gases.

For these applications involving parts not subjected to stress, corrosion resistance is the primary problem. But in structures that must carry high working loads



Relative cost of high-temperature strength is more for the high chromium alloys, limiting their use to conditions where corrosion is severe

the alloy steel must have good creep resistance as well as corrosion resistance at the operating temperature.

In equipment handling high-temperature steam, plain carbon steel is entirely satisfactory for load-carrying parts to 650 deg. F., but the strength of carbon steel falls rapidly at higher temperatures, as shown in the accompanying chart. Because of lower safe working stresses, carbon steel sections for high temperatures become so large that they are no longer economical when compared with more costly but stronger alloy steels.

For applications involving temperatures up to 950 deg. F., only small alloy additions greatly increase the creep resistance of carbon steel, as shown in the accompanying curves. For service at temperatures as high as 950 deg. F., the use of but ½ per cent molybdenum or 1 per cent tungsten increases allowable working stresses 2 or 3 times. The addition of nickel also increases the creep strength of carbon steel, 2 or 3 per cent nickel doubling the strength for temperatures up to 800 deg. F.

For still higher temperatures a 12 per cent chromium steel has a fair creep resistance and good corrosion resistance and can be used for stressed parts subjected to steam temperatures of 1,000 deg. F. The addition of about ½ per cent molybdenum or 8 per cent nickel to these higher chromium alloys further increases creep strength at higher temperatures and also gives an alloy that will not embrittle at temperatures up to 1,800 deg. F. For increased corrosion resistance at these higher temperatures, the chromium content is usually raised to 18 per cent or higher.

In many of these high temperature conditions, corrosion is just as important as high creep strength. Molybdenum and tungsten, although highly effective in increasing high-temperature strength add but little to corrosion resistance. In high-temperature service involving contact with flames, hot gases or heated corrosive liquids, other elements are needed in alloy steels for added resistance to attack.

As in steels for non-loaded parts, chromium is the most effective in increasing resistance to oxidation. In combination with ½ per cent molybdenum for added strength, 1 per cent chromium gives resistance to attack, in boiler tubes operating at 1,100 deg. F. and higher. Greater chrome additions provide increased resistance to the attack of more corrosive gases and fluids, or more active corrosion at higher temperatures, but add little to the creep strength.

Chromium for Corrosion Resistance

For service at still higher temperatures corrosion resistance is the primary problem. The addition of chromium in amounts up to 5 or 6 per cent makes a chemically stable alloy for temperatures of 1,200 deg. F. This alloy increases the life of boiler and superheater tubes to as much as 10 times the life of carbon steel tubes—more than enough to make the high cost alloy more economical.

In alloys for use in oil cracking stills the higher chromium content (4-6 per cent) provides resistance to corrosion from sulphur compounds, while ½ per cent molybdenum adds greatly to the creep strength. Although these properties depend somewhat on heat-treatment, service tests on these steels show that ductility and toughness are not impaired by extended operation at temperatures as high as 1,200 deg. F. A life of six times that of plain carbon steel is consistently obtained, more than making up for the difference in cost shown in the accompanying chart.

Because of their high cost, the straight 14 per cent chromium or 18-8 stainless steels are seldom used in high temperature service except where corrosion resistance is of primary importance. In the basic chrome-nickel-iron composition, chromium alone supplies the corrosion resistance. Nickel, although adding little to corrosion resistance provides the austenitic structure that makes the 18-8 stainless steel desirable for supporting high working loads.

QUESTION AND COMMENT

Pilot-Light Control

I. E. McCABE
Chief Engineer, The Mercoid Corporation

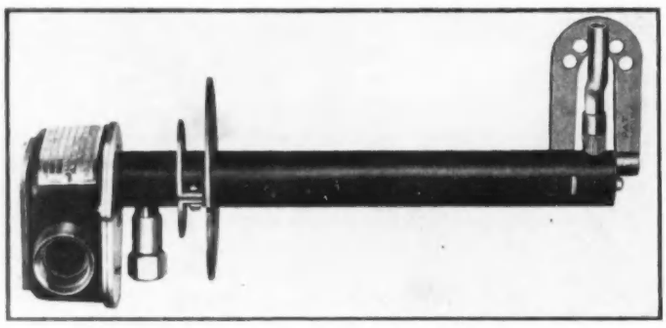


Fig. 1—Operation of the Powerstat pilot-light control is independent of surrounding temperatures

● For gas burners and gas-ignited oil burners, pilot lights must be protected so that failure of the pilot flame will prevent operation of the burner. Since the pilot light is in the flame zone the design of the control is complicated by the fact that it must be unaffected by surrounding combustion chamber temperatures. Such a mechanism, used in the Powerstat control shown in Fig. 1, operates on temperature difference rather than the actual temperature.

This device is actuated by a solid, single metal horseshoe element (not a thermostat) having five perforated holes located at the lower side of the bend. As shown in Fig. 2, under normal operation a small flame constantly heats the inside of the bend. Since the flow of heat to the outside of the bend is restricted by the holes, the expansion of the heated section opens the legs of the horseshoe element.

The outer prong of the element is pivoted in a fixed position at *M*. This is the end of the control that sets into the firepot of the furnace at a point where it can conveniently ignite the gas. The other prong of the element is locked in a rectangular hole of the rod *A*, at *C*, and is held under tension by means of the nut *B* at the other end of the rod. The tension is obtained by the outward pressure of two flat springs *G* and *H*, against which the nut *B* bears, acting as a knife hinge to prevent any binding action of rod *A* in its short travel.

When the pilot flame strikes the element, it immediately expands the perforated side and widens the gap between the two prongs, permitting the tension springs to move the rod *A* toward the stationary mercury switch. While the travel of the rod itself is small, the linkage gives a leverage of 6 to 1 between the rod and the permanent magnet, providing sufficient travel of the magnet to operate the mercury-enclosed contact switch. The magnet-carrying unit *L* is fastened to plate *F* which in turn is riveted to the tension spring *G*. The motion of the unit is confined between two adjustable set screws.

When the pilot is lighted and the horseshoe element expanded, the magnet is held close to the side of the glass switch, magnetic attraction holding the switch in the closed position. If the gas pilot is extinguished, the element quickly reaches a uniform temperature, and the contraction moves the magnet away from the switch, opening the electrical circuit. Since the speed of operation depends only on the rate of heat diffusion in the horseshoe element, the timing remains constant regardless of combustion chamber temperatures.

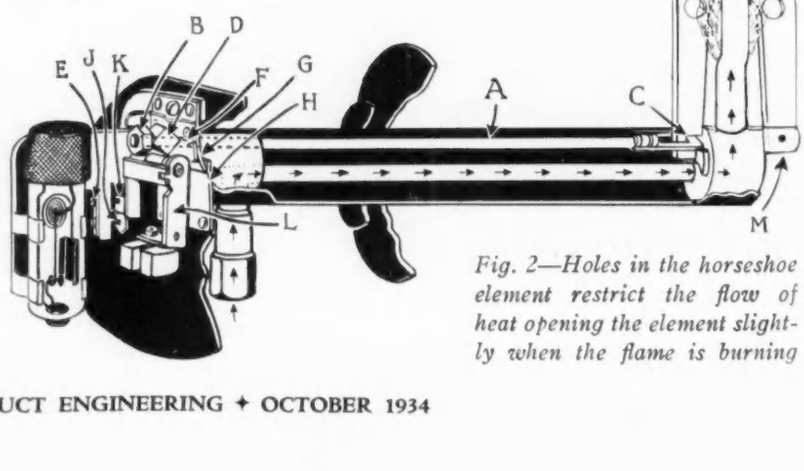


Fig. 2—Holes in the horseshoe element restrict the flow of heat opening the element slightly when the flame is burning

Using Consultants

To the Editor:

● In his article "Managing the Engineer" in the May number, page 186, M. G. Demougeot has a legitimate complaint, but his criticisms are not constructive. As I understand it, the machine to which he refers was probably completed in record time. Why should anyone assume that an engineer competent to design a machine is necessarily able to take care of all the small details, as that of ink in this case?

We have heard a lot about temperamental engineers, but it is easy to overlook the cricumstances under which engineers work. In a number of concerns it is customary, in instances like this, to engage outside engineering or consulting assistance to help the designing engineer and relieve him of any details he cannot handle efficiently,

either due to lack of time, or unfamiliarity with certain materials. There are a number of engineering organizations rendering such service and, only if the engineer refuses assistance are the remarks of Mr. Demougeot justified.

By nature management is conservative so as to maintain a balanced organization, but sometimes it becomes too conservative, ignores warring factions making no attempt to coordinate the different departments. It is generally true that the better organized a concern is and the better their product, the more they use independent engineering service to take care of their special problems. In my opinion the factory management should join the designing engineer in urging the engagement of experts competent to render such service. Thus costly delays to which Mr. Demougeot rightfully objects would be avoided.

—JOSEPH C. RAH
Chicago, Ill.

Plotting Curves

J. S. CARPENTER
Detroit, Mich.

● When plotting a curve on uniformly graduated cross-section paper with only a few points given, the points may not be close enough together to show the real trend of the curve. If the given points are first plotted on logarithmic paper, even if constants are present in the relation, the trend of the curve will be easily discernible. If the line so obtained is curved when plotted logarithmically, it can be assumed reasonably that the curve will pass through the given points. Inflection points present can then be fitted into the curve between the given points.

Curves so drawn will provide enough intermediate points of reasonable certainty so that the logarithmic curve can then be replotted on uniformly graduated cross-section paper. In the final curve unnatural "humps" and "dips" will be eliminated.

Internal Stresses in Die Castings

M. G. DEMOUGEOT
International Banding Machine Co.

● In a die-cast printing wheel having the complete alphabet cast on its outer circumference, there was introduced a difficult problem of dimensional changes caused by internal stresses rather than by aging, as experienced with some types of zinc alloys. The wheel, shown in the accompanying illustration, was 5 in. in diameter, with a cross section of approximately $\frac{11}{16}$x$\frac{1}{2}$ in.

Because of the design of the wheel, the casting could be gated at only one point, resulting in unequal and varying shrinkage. In order to obtain uniform castings, it is necessary to time the casting operation with a stop watch, allowing a uniform time of eleven

Courtesy New Jersey Zinc Co.

Simple heat-treatment relieved internal stresses in this die-cast printing wheel

seconds from the moment the die was filled with metal until the die was released. The operator removed each casting from the die, knocked off the gate, and immediately dropped it into a tub filled with running water. Within ten minutes after being cast, each die casting was checked for size and out-of-round, using a ring gage with a dial indicator, allowing a maximum total tolerance of plus or minus 0.003 in. on the diameter.

Just before machining, each casting was reinspected. Accurate measurement of the diameter showed that during the period of one to three weeks after being cast the die casting had shrunk uniformly about 0.003 in. To make all wheels uniform in shape and size they were then finish-bored on the inside diameter and the flat top of each letter was ground flat. Six months later, when again inspected, all wheels had shrunk an additional 0.001 to 0.002 in. and in addition were from 0.002 to 0.004 in. out-of-round.

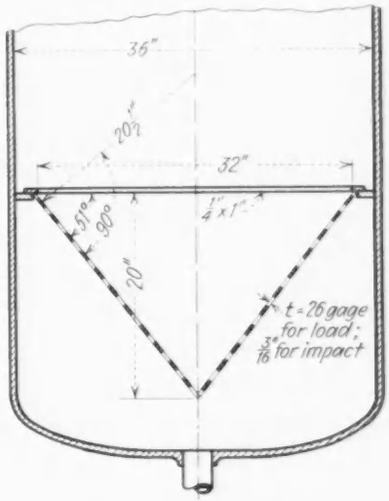
Since the Zamak 3 alloy used for this part is known to have high dimensional stability, internal stresses offer the only explanation for the shrinkage during the six months after casting. The parts had a hard, chilled outer surface containing a high degree of internal stress. Upon removing all of this hard, chilled surface when boring the inside of the wheel, and only removing a small portion of the chilled surface on the outside, a comparatively high stress was imposed on the core area, thus causing the wheel to gradually contract until the internal stresses were equalized.

Experience with this design showed that most of the internal stress, and therefore most of the shrinkage, could be eliminated by a simple heat treatment at a temperature of 160 to 200 deg. F. for 3 to 12 hours.

Perforated Cone Design

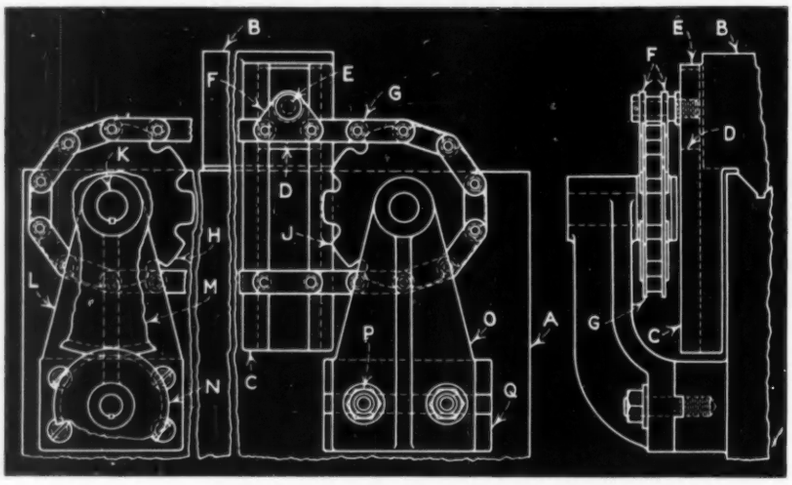
To the Editor:

● In the August number of *Product Engineering*, page 308, E. A. Tschopp requests help in the design of a perforated cone, or hopper, supporting a burden of broken stone.



Part of the column of broken rock is supported by friction against the walls, the cone supporting only a column 96 in.

Tests of deep grain bins have disclosed that when the depth of grain exceeds three diameters there is no increase in the weight carried by the bottom—the additional load being transferred to the walls by friction. In the present case the 2-in. ledge formed by the flange of the cone produces a corbelling effect and further reduces



Low-cost chain drive provides a long reciprocating motion without shock at the end of the stroke

the load carried by the cone. It can be assumed then, that the maximum load carried by the cone is a column of stone 32 in. diameter and 96 in. deep, or 4,500 lb. if the stone weighs 100 lb. per cu.ft.

The maximum stress in the cone, both tangential and meridional, is at its base. The angle of the side of the cone with the horizontal is nearly 51 deg. and the normal radius of the base of the cone is about 20½ in., as shown in the figure.

The meridional stress is the load times the cosecant of the angle 51 deg., which is 4,500 × 1.2868 or 5,790 lb. Dividing by the periphery, the load is 5,790/32 π or 57.7 lb. per linear in. The stress in a longitudinal seam of a cylindrical pressure vessel is twice that in a girt seam. Likewise, the tangential stress in a conical bottom or head is twice the meridional stress. The maximum tangential stress in the present case is 2 × 57.5 or 115 lb. per linear in. The same result is obtained directly by multiplying the pressure in lb. per sq.in. by the normal radius of 20.5 in.

With ½-in. holes on 1-in. centers the efficiency of the plate is 50 per cent, assuming that the welded seams avoid the holes. With a unit stress of 12,000 lb. per sq.in., the required thickness of sheet for a static load is 115 (0.50 × 12,000) or 0.0192 in., say 26-gage.

If the cone merely rests, by its flange, on the supporting ring, then provision must be made to resist the horizontal component of the meridional forces in the cone. The total meridional force was found to be 5,790 lb. Dividing by 2 and using a working stress of 12,000 lb. per sq.in., the required cross-sectional area of the flange is 5,790 (2 × 12,000) or 0.241 sq.in. Of course, the 26-gage metal of the cone would not do for the flange, making necessary a bar ¼ by 1 in. to which the cone can be welded as shown in the figure.

The cone just described, although capable of carrying the static load, is not adapted to resist the impact of falling stone. If the container is charged from the top, and the stone allowed to fall freely, the cone must resist the impact resulting from a fall of about 14 ft. Obviously, 26-gage metal will not last long under such treatment, so the cone should be made of material at least ⅜ in. thick with the edge flanged to a width of 1.5 in., or welded to a ring as in the figure.

—C. O. SANDSTROM
Thermal Engineering Company

Practical Professors Needed

To the Editor:

● With reference to the article "Learning Rules and Rote" that appeared in the September number, I agree with your managing editor that college courses in machine design certainly do not reflect the problems of the designing engineer as he meets them in industry.

Looking back ten or fifteen years upon my own college course, I can distinctly remember our professor in machine design lecturing on the question of selection of material. The keynote of his instruction was that, because of its cheapness, cast iron is generally used in preference to steel, or other metals. His theory seemed to be that if cast iron could meet the strength requirements, it was the best material to use.

Of course, it is absurd, but if we follow such an argument to its conclusion, cast iron would be practically the only material in commercial use. This particular professor had no conception of the basis for selecting a material except that of stress and strain. Required surface finish, corrosion resistance, weldability, weight-strength ratio, pattern and tooling costs and the many other similar factors that govern the selection of a material, were not even mentioned in his lectures. The limits of his knowledge and experience seemed to be that of equating unit strength with unit stress.

From my discussions with other engineers who have been through technical colleges, I am convinced that this is quite a general situation. I would suggest that our engineering colleges appoint to their faculties designing engineers of experience to teach the subject of machine design. It may be that such men would be short on the academic side, but that should not be given too much consideration. Technical courses should be taught by experienced technical men rather than teachers of theoretical mechanics.

—H. E. TAYLOR
New York City

Long Reciprocating Movement

J. E. FENNO
Belleville, N. J.

● Wood planer knives are ground all over and are usually given a final polishing operation on a surface polishing machine. These blades, sometimes 10 ft. in length, require a long reciprocating table movement to carry the polishing head across the length of the blade. For a low-cost, adjustable drive, the simple sprocket and chain design shown in the accompanying diagram provides a smooth reciprocating drive for the polishing head, and also provides a gradual retardation and acceleration at the reversal of direction, eliminating shock at the end of each stroke.

Cast integral with the left-hand side of the table *B* is the apron *C* which has a vertical guide for the sliding block *D*. To the stud *E* in this block is pivoted the triangular link *F* which forms one of the links of the endless chain *G*, passing over the sprockets *H* and *J*. Gears *M* and *N* drive sprocket *H* at a constant speed, while the driven sprocket *J* is provided with a horizontal adjustment to take up wear in the chain.

As indicated by the arrow, the upper side of the chain moves to the right, carrying the table with it in this direction. When the link *F* moves downward around the sprocket *J*, the move-

ment of the table is gradually retarded, further rotation reversing the direction of table travel.

Design of Perforated Filter Cone

To the Editor:

● Mr. Tschopp's question on page 308 of *Product Engineering* for August, 1934, needs further data. To apply the following approximate method, the specific weight of the broken rock based on occupied volume and its angle of repose as installed in the shell are needed and should be experimentally determined.

The cone is not hydrostatically loaded, but supports the weight of its own contents or the "contained" material, and that of a superimposed conical heap or the "piled" material. The rest is taken by friction on the shell and by the internal supporting flange. The weight of the piled material varies with the specific volume as installed and the angle of repose.

An approximation of the stress may be made by assuming that the loading is hydrostatic and that the cone is not perforated. This analysis gives the approximate stress for unit thickness of material, and out of this a suitable thickness may be obtained to satisfactorily meet the yield strength and the safety factor for an imperforate plate. But the presence of the holes makes the problem indeterminate, and introduces stress concentration. Therefore some judgment must be exercised in allowing for these two effects. The thickness as determined for the imperforate cone can be multiplied by the ratio of pitch of perforations to the hole diameters, and by a stress concentration factor of say at least two. The final thickness is then at least four times that of the imperforate case.

This procedure gives one form of mathematically directed guess as to the desired dimension. Tension and shear at junction of flange and cone may be checked in the customary manner, if this region is suspected.

—Roland V. Hutchinson
Lansing, Mich.

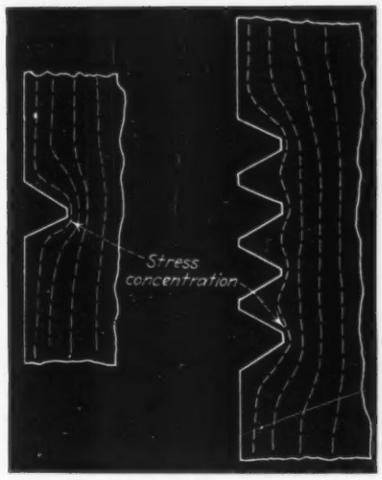
Stresses in Threads

To the Editor:

● With reference to Prof. H. F. Moore's discussion on p. 307 of the August number, it is true that fatigue tests on metal models, say standard test pieces, show a lower stress concentration factor than obtained from photo-elastic experiments. That is, it seems that fatigue specimens show in the laboratory a higher strength than anticipated from photo-elastic studies. One reason is the yielding of metal, or the "slight plastic action" mentioned by Professor Moore. Another reason is the unavoidable discrepancy between a flat test piece as used in photo-elasticity and a three-dimensional test piece such as a rotating beam or a threaded bolt.

The extreme example of increased strength caused by three dimensional conditions is the inner (or outer) race of a ball bearing. The stresses actually existing under the ball are of the order of 100,000 to 250,000 lb. per sq.in., yet the metal does not fail. The metal "has nowhere to be squeezed out" and therefore does not give in to the stress.

In the flat photo-elastic test piece there is nothing to restrain the metal



Stress concentration is much higher at the bottom of a single notch than at the bottoms of a series of notches

from increasing its thickness where it is compressed or from decreasing its thickness where it is in tension. Photo-elastic investigations deal with two dimensional models only, and a correction must be introduced when the observations are later applied to three-dimensional stress conditions.

Professor Moore's comparison of the coefficients found by Hall and Solakian is interesting. The concentration of stress is much higher around a single notch (a thread is to a certain extent a negative notch) than around a series of notches, as shown in the accompanying sketch. The lines are more distorted around the single notch, and therefore there is more concentration of lines near the bottom of the single notch. This in itself should be sufficient reason for Hall's factor of 5.6 as compared with the factor of 3.9 obtained by Solakian.

—G. B. Karelitz
Columbia University

Stresses in a Large Spring

To the Editor:

● In calculating the working stresses for the enormous spring shown in the middle of page 23 of the advertising section of your August number, the chart given on page 280 of the July number indicates a deflection per coil of 1.9 in., based on a closing load of 70,000 lb. The working stress is then 175,000 lb. per sq.in. However, the picture of the spring being compressed shows 9 turns, and using 9 for the number of active coils, the spring deflection is 1.64 in. per coil and the load required to compress the spring is then 60,400 lb., instead of 70,000 lb., as stated. The working stress is 151,000 lb. per sq.in., which is still high for stock 2.5 in. in diameter.

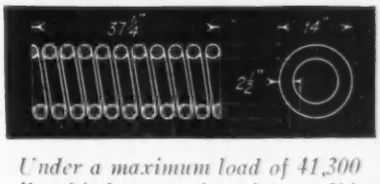
Calculating its length from the volume using the weight of 500 lb. as given, the spring has 10 turns. The deflection per coil is then 1.225 in. for complete closure of the spring and the working stress is 110,000 lb. per sq.in. Under these conditions, the load required to close the spring is 45,100 lb., but the caption at the bottom of the page states that the closing load is 70,000 lb.

—Carl P. Nachod
Louisville, Ky.

To the Editor:

● Design dimensions of this spring are shown in the accompanying diagram. The solid height is 26¼ in. with 10½ total turns, and a bar length of 379 in. before tapering. Using a torsional modulus of 10,500,000 and considering 1½ turns inactive, the space between turns is 1.222 in., giving a load capacity of 41,300 lb. when compressed solid. This load corresponds to a maximum fibre stress of 77,500 lb. based on standard formulas or 103,800 lb. per sq.-in. when corrected by the Wahl factor.

Although the maximum capacity of a spring when compressed solid is



Under a maximum load of 41,300 lb., this large spring closes solid, the maximum fiber stress being 103,800 lb. per sq.in.

41,300 lb., the 70,000 lb. load was applied to lessen winding kinks slightly and as a further test of the material.

—DuRay Smith
General Manager
Union Spring & Mfg. Company

Eight Types of Retaining

Eighteen applications of pin locks and six spring locks.

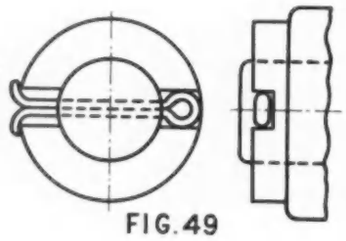


FIG. 49

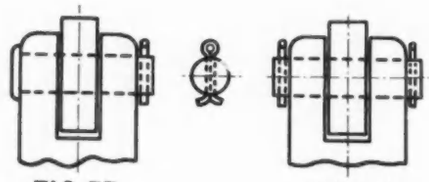


FIG. 53 FIG. 54

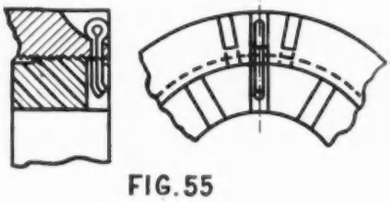


FIG. 55

ADAM FREDERICKS

Figs. 49 and 50—Cotter pin in collar slot prevents rotation.

Fig. 51—Loose pin in spring actuated rod is encased in cup washer for retention.

Fig. 52—Extension spring anchor and method of retaining the anchor pin.

Figs. 53 and 54—Cotter pins retain roller shaft.

Fig. 55—Cotter pin retains threaded collar having milled slots, for close positioning.

Fig. 56—Cotter pin holds castellated and slotted nuts.

Fig. 57—Spring backed lock plunger is retained by spinning over hole.

Fig. 58—Two pins with bevelled ends engage internal vee-groove, actuated by cone pointed screw.

Fig. 59—Locking plunger with handle attached.

Fig. 60—Spring backed plunger with hemispherical end engages vee-slot.

Fig. 61—Flat spring retains two plungers.

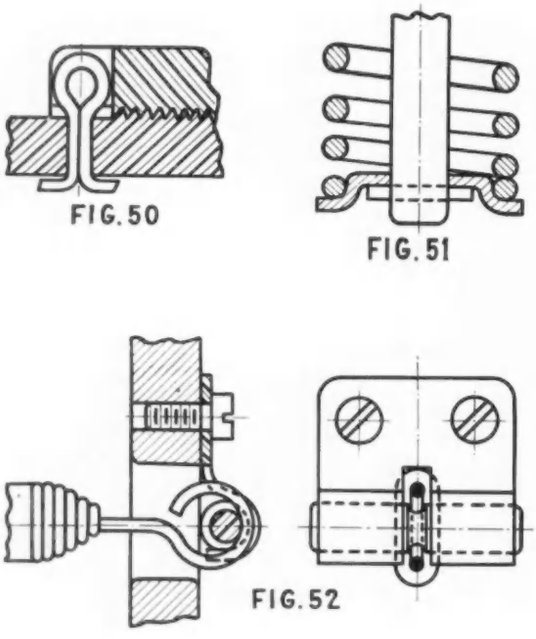


FIG. 50 FIG. 51

FIG. 52

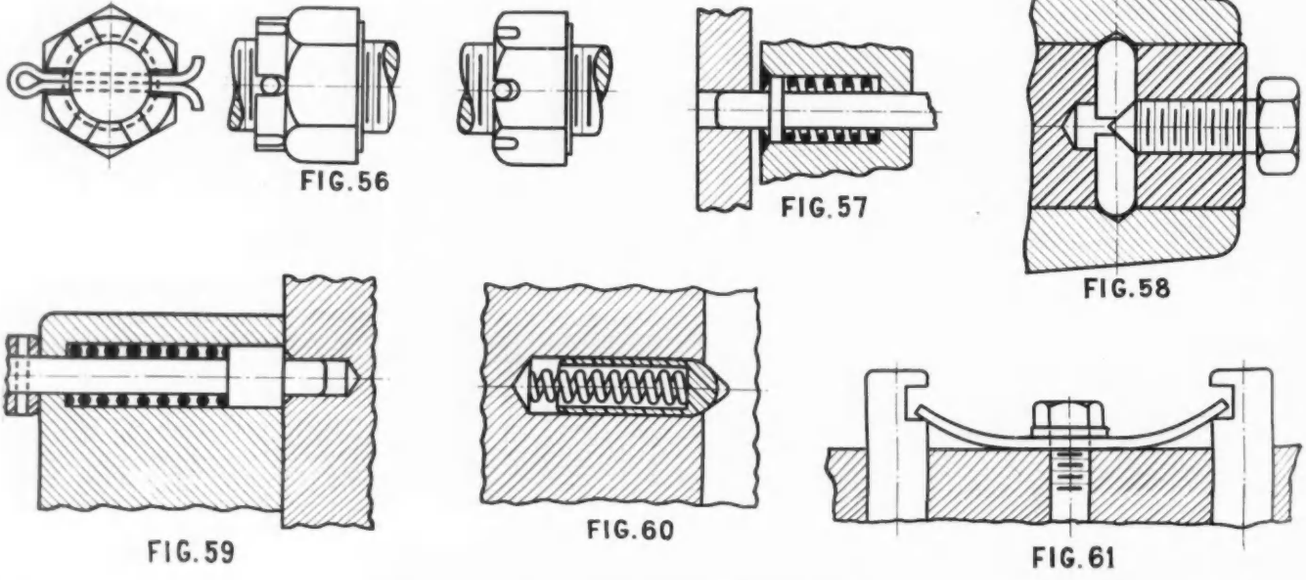


FIG. 56 FIG. 57 FIG. 58

FIG. 59 FIG. 60 FIG. 61

and Locking Devices—III

The concluding article will appear in the December issue

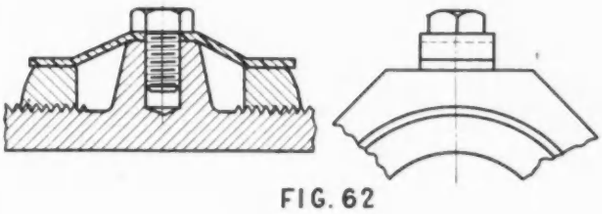


FIG. 62

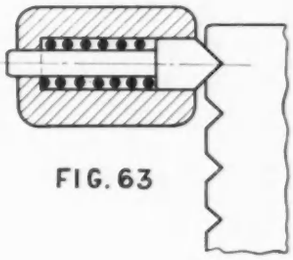


FIG. 63

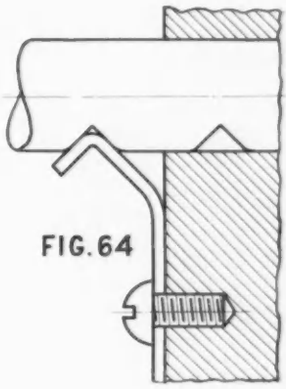


FIG. 64

Fig. 62—Flat spring contacting flats of two hexagon nuts prevent backing off.

Fig. 63—Locating plunger having no retention.

Fig. 64—Flat spring formed with hump to engage vee-slots in sliding rod.

Fig. 65—Flat spring with rounded head *A* is rivetted to pin *B*. By compressing spring, pin is removed with knurled knob.

Fig. 66—Spring-backed ball engaging vee or half-round groove, retained by headless screw.

Fig. 67—Oversized slotted plug, when pushed in hole, presses the four segments outwards.

Fig. 68—Formed wire with pin end engages holes in nut and shaft keyway.

Fig. 69—Single coil of compression spring resting in collar groove locks screw.

Figs. 70 and 71—Single coil of spring retains plug in hole.

Fig. 72—Snap ring in groove of pin is retained in counterbored hole in inner member.

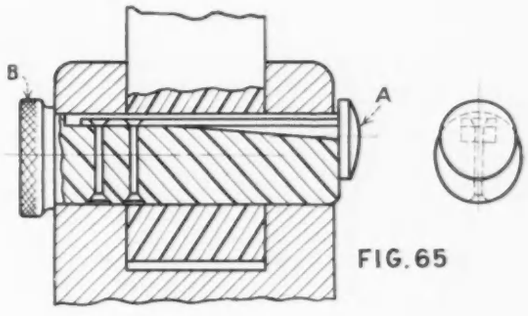


FIG. 65

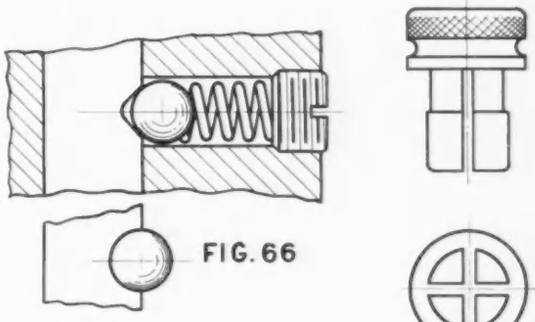


FIG. 66

FIG. 67

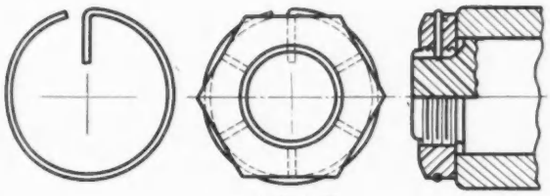


FIG. 68

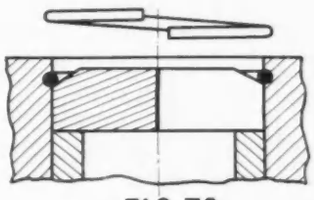


FIG. 70

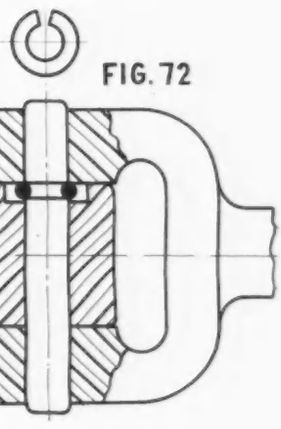


FIG. 72

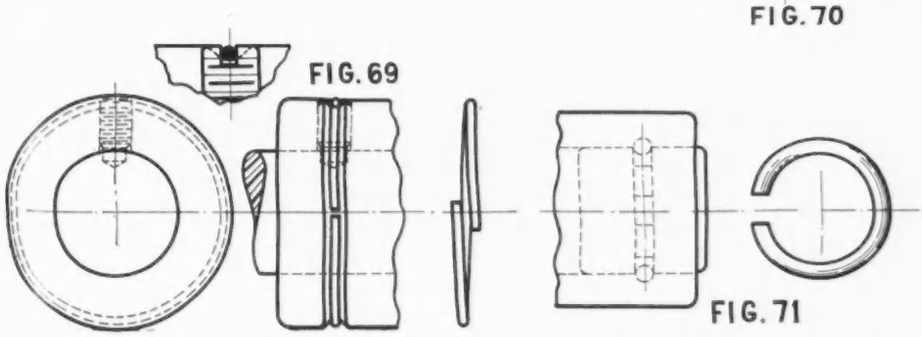


FIG. 69

FIG. 71

NEWS

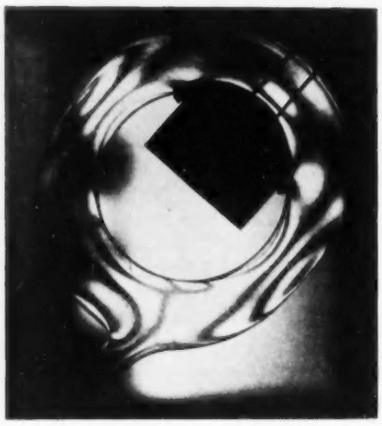
Photo-Elastic Test Used as Final Check

IN THE design of a yoke for a large and heavy bending roll, engineers of the Baldwin-Southwark Corporation were faced with an intricate problem in stress distribution. The only feasible construction was to use a casting made of high tensile electric furnace steel and it was estimated that it would weigh about 11,000 lb. The casting would be called on to safely resist direct loads up to 1,650,000 lb. normally and probably more than twice this load occasionally. At the same time the parts would be subjected simultaneously to side loads of smaller magnitude but producing a transverse bending in a vertical plane. Because of the shape of the part, together with the manner in which it would be loaded, it made analytical investigation difficult, if not impossible.

In order to determine the magnitude of the stresses that would be developed in the casting, the engineers decided to make a scale model out of steel plates, following the well-known laws of similitude and loading it in a testing machine, and measuring the strain with the Huggenberger Tensometer. This was done, and from the measured strain the stresses were readily calculated.

However, this method was not entirely satisfactory because some of the results did not seem to be consistent with others. It was suspected that there were rapid stress changes in certain portions of the casting. Because the Tensometer had a minimum gage length of 1 in., it was impossible to detect such sharp stress gradients which were expected to exist in several parts of the cross section.

It was decided, therefore, by the engineers to pursue the matter further and investigate the stress distribution by photo-elastic methods. Since no photo-elastic outfit was conveniently available, it was decided to build the necessary equipment. Since the primary purpose was the detection of dangerous stress concentration, if any, and not a quantitative analysis, the problem was rendered relatively simple.



In the accompanying illustration is shown a view of the stressed model viewed in panchromatic polarized light. In this instance, the results showed, as may readily be seen in the illustration, that no dangerous stress concentration was present, and that the part was designed with a reasonable assurance that, given a normal casting, no field failure was probable.

Color Control for Modern Finishes

INDICATIVE of the generally recognized facts that color contributes greatly to the sale value of a product, manufacturers in various industries are now giving the color problem more attention. It is not sufficient that the engineer specify certain colors to be used. It is necessary that in the production of the product the color applied must be as specified.

For a number of years the Packard Motor Car Company has been using the color analyzer to enable them to reproduce exactly the colors specified.

A more recent development is the color matcher developed by Westinghouse Electric & Manufacturing Company to match porcelain panels for refrigerators made at their Mansfield Works. This color matcher operates by means of a photocell. It quickly and accurately selects the panels that match.

In the operation of this color matcher, the colored light is sent to a sample and the light that is diffusely reflected is sent to a photo-tube controlling an amplifying tube and an indicating microameter. The colored light is provided by three color filters, red, green and blue. Rays of light pass through the color filter, lenses, 45 deg. mirror, and then through the window to the sample. A sample having a glossy surface reflects the ray back to the mirror and through the system back to the lamp. The true color of the sample diffuses the light, reaching the photo-tube by multiple reflection in the integrating chamber. In this way the relative readings are obtained on a meter and are compared with those of a standard sample.

Plans for Dark Rooms

INDICATIVE of the increasing extent to which investigators are resorting to photomicrographic studies, the Bausch & Lomb Optical Company received so many inquiries for assistance in laying out photomicrographic dark rooms, that they have prepared suggested plans for such rooms. One of these plans calls for the smallest usable space, 6 ft. x 6 ft., for which any small room or closet, or even a corner of a room, may serve the purpose. The other plan is for an average size dark room, 9 ft. x 12 ft.

These layouts show the suggested arrangements of sink, lights, shelves for chemicals, plates, films, paper, table, print-washer and drain-boards. Estimated costs for carpentry, painting, plumbing and electrical wiring are given. The customary materials needed for the dark room are listed and their total cost estimated.

The plans are printed on one side of a heavy cardboard poster, 9½ in. x 13 in., on the other side of which the rules for using the dark room are printed, with a bold caption, "Knock Before Entering."

The Bausch & Lomb Optical Company have announced that copies of these plans may be obtained on request.

Welding Clinic Announced

THE fourth welding clinic sponsored by the Linde Air Products Company, has been announced to be held in New Orleans starting October 24, and continuing to October 27.

The express purpose of these clinics is to give to those interested in oxy-acetylene welding and cutting informa-

tion on the new developments in the process; to give assistance to those who have welding and cutting problems; to demonstrate and give instructions on new and old applications of the oxy-acetylene process; to afford a chance for discussion and interchange of ideas among those attending the meeting.

The New Orleans Clinic will be held at the Isaac Delgado Central Trade School, the officials of which are offering all the facilities of the school to take care of the large attendance. Complete programs have not been announced, but will include demonstrations of several new developments in welding and cutting equipment and both usual and unusual applications of the oxy-acetylene process. There will also be informal discussions, motion pictures, exhibits of unusual and standard welding and cutting jobs, and testing of coupons welded by the guests.

Aging of High-Carbon Steel

NEW methods of thermomagnetic analysis, or measuring magnetization continuously during heat-treatment, show that even at room temperature, quenched steel undergoes progressive changes. The effect of these changes is to produce a decrease in hardness, rather than the increase found in "age hardening" phenomena, according to G. A. Ellinger and R. L. Sanford in Research Paper RP 707, appearing in the National Bureau of Standards *Journal of Research* for August.

Changes occurring during prolonged tempering at 200 deg. C. are attributed to a relief of internal stresses set up by phase changes taking place during quenching. At 300 deg. C., tempering had little effect upon the changes occurring, while aging at room temperature increased both the magnitude of the changes and the time required.

Design Papers at A.S.M. Meeting

NEARLY every session scheduled by the American Society for Metals at their annual meeting in New York during the week of October 1 to 5, includes papers of interest to designers. Among these papers are reports on: Physical properties of case hardened steel in the heat-treating session; properties of cast alloy steels, corrosion resistance of wrought iron, and oxidation resistance of low carbon steel at high temperatures, in the research sessions; directional properties in rolled and annealed low carbon steel in the session on aging of steel. Other papers of special interest will give results of tests on the ductility of stainless steels at elevated temperatures, and the impact resistance of hardened carbon steels.

As announced here in the August and September numbers, many sessions scheduled by other societies cooperating in the general program for the National Metals week also promise to have much of direct interest to designers. At the Exposition, many action displays, working models of metal processes, and exhibits featuring welding are being prepared.

MEETINGS

National Metal Congress — New York, N. Y., Oct. 1-5, the following societies participating:

American Society for Metals—Hotel Pennsylvania, Oct. 1-5. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.

American Welding Society—Hotel New Yorker, Oct. 1-5. Miss M. M. Kelly, secretary, 29 West 39th St., New York, N. Y.

American Institute of Mining and Metallurgical Engineers — Institute of Metals Division, Hotel Pennsylvania, Oct. 2-4. John T. Breunich, assistant secretary, 29 West 39th St., New York, N. Y.

Printing Research Institute — Joint meeting with the A.S.M.E. Printing Industries Division, Benjamin Franklin Hotel, Philadelphia, Pa., Oct. 8-9. E. P. Hulse, secretary, 29 West 39th St., New York, N. Y.

American Foundrymen's Association—Fifth international congress and 38th annual convention, Convention Hall, Philadelphia, Pa., Oct. 22-26. R. E. Kennedy, secretary, 222 W. Adams St., Chicago, Ill.

EXHIBITIONS

National Metal Exposition—Port Authority Bldg., New York, N. Y. Oct. 1-5. W. H. Eisenman, director, 7016 Euclid Ave., Cleveland, Ohio.

Brewery Supply and Equipment Exposition—Grand Central Palace, New York, N. Y., Oct. 8-13. Felix Mendelsohn, director, Hotel Lexington, New York City.

Industrial Materials Exhibit — Park Central Hotel, New York, N. Y., Oct. 15-19. W. Wyler, manager, 484 Broome St., New York, N. Y.

National Business Show—Annual exhibit, Commerce Hall, Port Authority Bldg., New York, N. Y., Oct. 15-20. F. E. Tupper, managing director, 50 Church St., New York, N. Y.

American Foundrymen's Association—Exhibition of foundry equipment, Convention Hall, Philadelphia, Pa., Oct. 22-26. C. E. Hoyt, manager, 222 W. Adams St., Chicago, Ill.

Power Show—11th National Exposition of Power and Mechanical Engineering, Grand Central Palace, New York, N. Y., Dec. 3-8. Charles F. Roth, director, Grand Central Palace, New York, N. Y.

Higher Die-Casting Pressures

FURTHER developments in die-casting equipment and procedures now permit die-castings to be made under pressures as high as 3,000 to 17,000 lb. per sq.in., according to N. N. Lester, chief engineer of the Pressure Castings, Inc. This has been made possible largely by the development of a new type of casting process wherein the pressure is applied hydraulically.

In this high-pressure casting process, a charge of the molten metal is either automatically or manually picked up and placed into a cylinder in close proximity to the die. Here it comes into momentary contact with the cold surfaces of the cylinder and plunger head, the resulting chilling causing a thin, protective shell of solid metal enveloping the molten metal.

Almost simultaneously, the plunger is actuated. This crushes the thin solidified shell and propels the molten metal under great pressure into the die cavity. During the fraction of a second in which the die-casting plunger is being moved by the hydraulic cylinders to inject the metal there is a complete scavenging of the air from the mold. It is claimed that this insures perfect solidity, homogenity and density in the resulting die casting.

This hydraulic die-casting method is being applied to castings of aluminum, brass and bronze, particularly where thin metal walls are required. It is claimed that such high-pressure die-castings have greatly increased tensile strength because of their greater density.

New Materials and Parts

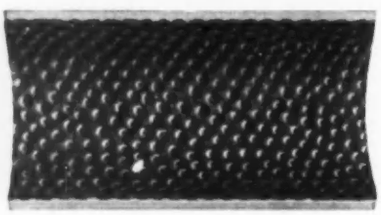
High-Speed Welding Electrode

Designated as Type W-23, heavily-coated electrodes for arc welding are for use on high-speed welds in the flat position. For either manual or automatic arc welding, electrodes are said to produce good results on either alternating or direct current. Typical physical characteristics after stress relieving at 1200 deg. F. are:

Ultimate tensile strength, lb. per sq.in.	66,000
Yield point, lb. per sq.in	54,000
Elongation in 2 in., per cent	32

Electrodes recommended for fabrication of pressure vessels, pipe, machinery, and other joints where welding can be carried out in the flat position. General Electric Co., Schenectady, N. Y.

Electrunite Conduit



Tubing for inclosing electrical wiring is provided with an internal surface covered with small, round, raised knobs. This surface is claimed to reduce surface friction and jamming in pulling cables through the tubing. Three fittings are said to adapt this conduit to any job, eliminating threading. Tubing is also said to be easier to cut, bend and straighten. Steel and Tubes, Inc., 224 East 131st St., Cleveland, Ohio.

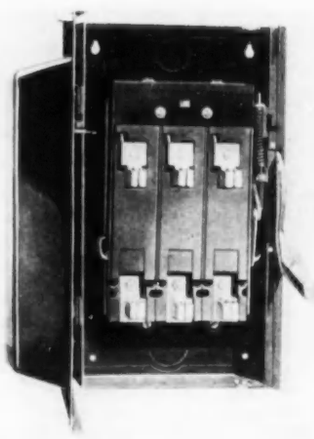
Automatic Motor Starter

Prevents accidental starting and guards against overload. Red stop button extends around and beyond the green start button; once stopped, motor can be restarted only by pressing the recessed start button. Inverse time limit protection limits overload. Low-voltage release with two-wire

control insures immediate motor starting upon resumption of current. Available in three types in various ratings, and for 110, 220, 440 and 550 volts. Mechanical moving parts are cadmium plated. Moving contacts can be removed without tools; stationary contacts removable by use of a screw driver. Mechanism can be installed and replaced without disturbing conduit, a single screw holding the switch panel to the back of the case. Lincoln Electric Co., Cleveland, Ohio.

High-Capacity Motor Switches

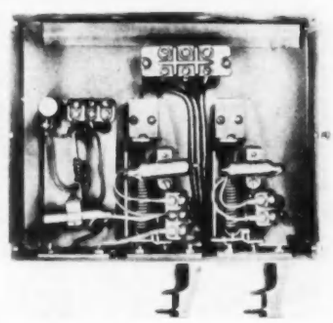
Available in either standard sheet metal boxes or cast aluminum enclosures, with cast iron cases optional. Switches of both 100 and 200 amp.



capacity are quick-make, quick-break, with interlocked covers and elevated removable bases for easier wiring. Double-break contacts have steel spring reinforcement to insure constant pressure on the contacts. Available in two and three poles, 3-wire solid neutral and 4-wire solid neutral, fusible and not fusible. Cast enclosures are weatherproof or dust tight. Square D Co., Detroit, Mich.

Automatic Pump Controller

To maintain water level in an elevated tank or standpine between maximum and minimum levels. An electrical "snubber" or time delay element eliminates surging effect. Snubber consists of a bimetallic strip wound with a heating element which controls the stop switch. Two pres-



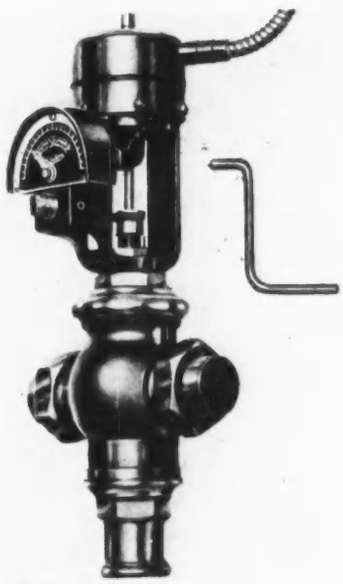
sure bellows actuate mercury switches to make contact below low level and above high level. High level switch energizes heating element, opening snubber switch and stopping the pump. Time delay adjustable. Cover, top and bottom plates are removable, all equipment being attached to the bottom plate. Switches rated at 10 amp., 110 volts; 5 amp., 220 volts. Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

Cork Sheet

Consisting of a thin coarse-mesh fabric reinforcement faced on both sides with cork, bonded together through the cloth. Composition is said

to have the same sealing and cushioning properties as those of ordinary cork composition, but possesses four times greater tensile strength. Supplied in standard thicknesses of 3/32, 1/16, 1/32 and 1/8 in., in continuous rolls up to 72 in. in width. Armstrong Cork & Insulation Co., Lancaster, Pa.

Adjustable Temperature Regulators



Graduated dial and pointer, adjusted by a removable crank, changes setting of regulator, one turn of the crank changing the setting 10 deg. F. Regulator operates through a standard temperature range of 100 deg. F. Regulator operates on vapor expansion principle, driven by an all-metal seamless bellows. Available in six standard temperature ranges, with space available on the dial for special marking. Fulton Sylphon Co., Knoxville, Tenn.

Free-Machining Screw Stock

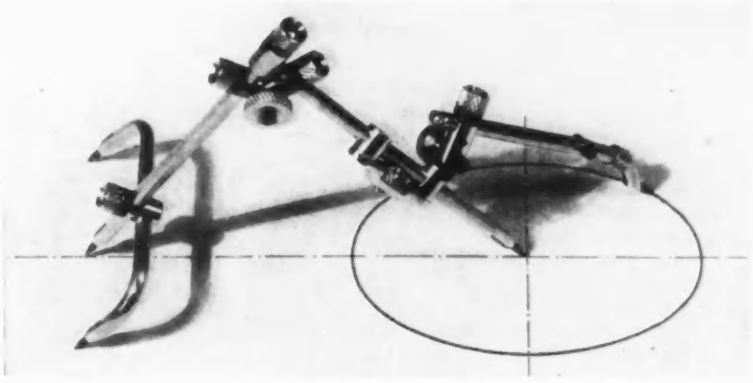
Screw steel of improved machinability obtained without changing analysis or altering physical properties. Manufacturing process is said to largely eliminate abrasive elements which tend to wear away cutting edges of tools. No higher feeds and speeds are claimed, but tool life is said to be extended from 100 to 400 per cent. New process is standard for all three types of free-machining grades. Union Drawn Steel Co., Massillon, Ohio.

Alloy Cast Iron

Made possible by a recently developed air furnace, a series of nickel-molybdenum iron alloys designated as Cecolloy are available for large castings. Alloy is said to be suitable for anvils and frames in forging hammers and other heavy machinery, combining the mass and vibration-resisting properties of gray iron with the strength and wear-resisting properties of steel. Tensile strength is said to be from 40,000 to 60,000 lb. per sq.in. with a Brinell hardness that can be controlled in the furnace as required. Carbon contents can be held within plus or minus 0.05 per cent. Material is said to finish with a clean, smooth surface because of its fine, graphite-free grain structure. Chambersburg Engineering Co., Chambersburg, Pa.

Elipsograph

For drawing any ellipse up to a major axis of 5 in. and a minor axis of 3 in. Ratio between axes adjusted by varying the distance between the 3-point support and the intersection of the axes, a greater distance increasing the ratio. Pencil arm adjustable for the desired minor axis. One half of the ellipse is drawn from each side of the vertical center line, the compass unit being free to slide on the leg resting on the centerline intersections. Instrument assembled from parts included with drafting tool announced on page 357 in the September number. R. B. Ware, 98 Holmes St., North Quincy, Mass.

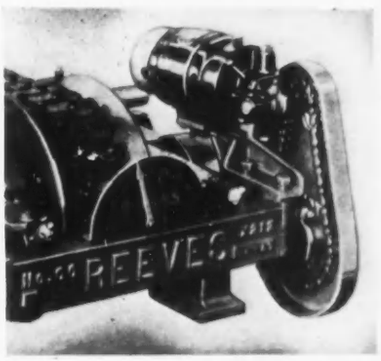


Exhaust Valve

Said to give four to five times longer life than valves made of plain steel. Is constructed of a molybdenum steel forging with a seating surface of

a silicon-chrome steel alloy, deposited by a special welding process. Available in sizes up to 3 in. in diameter and larger. The Meriam Co., Cleveland, Ohio.

Motor Control for Variable Speed Unit

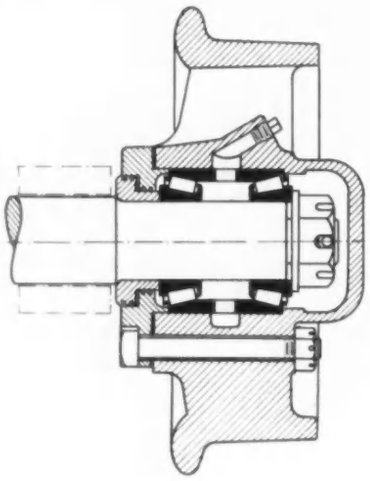


For textile, paper, wire, veneer, ceramic and machine tools, an attachment for the Reeves variable speed drive permits automatic and remote adjustment of the variable speed drive. An a.c. or d.c. electric motor with built-in speed reducer and a safety clutch to protect the motor against overload, sets the drive to the required speed ratio. Output shaft of the motor-reducer running at 150 r.p.m., drives the transmission shifting screw at a 3 to 1 ratio, through a roller chain. Motor is rated at 1/4 hp. for either 100-125 volts or 200-250 volts. Control available for installation of units already in service, either vertical or hori-

zontal type; and can be supplied integral with the more generally used sizes. Reeves Pulley Co., Columbus, Ind.

Double-Cup Bearing

Designed to eliminate loss of lubricant or entrance of dirt and water, in mine car wheels or construction ma-



chinery, making removal of wheels possible without disturbing bearing. Wheel hubs can be bored in a single operation without shoulder. Bearing remains on the axle as a complete unit, the hub being machined for a slip fit over the bearing cup. Design can be readily adjusted for wear. Bearing takes both radial and thrust load. Timken Roller Bearing Co., Canton, Ohio.

Silverstreak Silent Chain

Special finish available for all standard types of chain. Washers and guide bars with a silvered appearance and the remainder of the chain blued. Silvered parts thus appear on a dark blue background, adding a distinctive appearance. Link-Belt Co., 910 South Michigan Ave., Chicago, Ill.

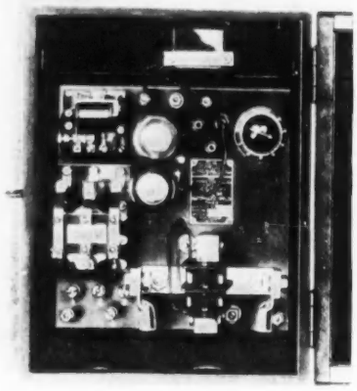
Small Blueprinter

Complete cabinet blueprinting equipment stored in cabinet occupying only 18x22 in. floor space. Side leaves of all-steel cabinet when raised accommodate trays and dryers. Drawer has two light-tight compartments. Projector, made of pressed steel, is fitted with an arc lamp and an automatic timer which rings bell when exposure

is completed. Resistance unit for arc lamp mounted on back of printer. Dryer designed to dry print in one minute without scorching. Makes prints up to 9x12 in. F. A. Smith Manufacturing Co., Inc., Rochester, N. Y.

Automatic Weld Timer

For use with resistant welding machines, times the weld in inverse proportion to current flow. Inclosing cabinet is dust-tight, is smaller than previous designs and contains less apparatus. High speed of this timer is said to make possible the use of higher currents for welding materials of high heat conductivity. Timer incorporates three current ranges selected by inserting a plug into one of three re-



ceptacles on the panel. The Electric Controller & Mfg. Co., 2702 East 79th St., Cleveland, Ohio.

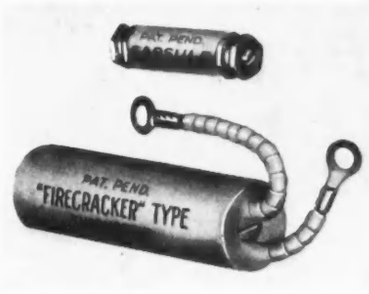
Immersion Heaters

For such applications as coffee urns or food process heaters, unit is attached through a hole in the bottom of the tank. Fitting is supplied with rubber gaskets for water tight joints. Available in 8 capacities from 880 to 6,000 watts, for 115 or 230 volt supply. Unit supplied with flexible cord connectors; three-heat switches available. Harold E. Trent Co., 618 North 54th St., Philadelphia, Pa.

Bimetal Thermostat

Entirely sealed and dustproof, thermostat is said to permit safe operation even in combustible gases. Electrolytic silver contacts are used. Metal plug with a headless self-locking stainless steel adjustment screw gives operation at temperatures adjustable up to 350 deg. F. Pig-tail terminals are insulated with ceramic beads. Small capsule type illustrated is $1\frac{1}{4}$ in. long, $\frac{5}{16}$ in. in diameter, rated at 100 watts a.c., or 50 watts d.c. heater load at 110 or 220 volts. Tem-

perature differential is 2 deg. F. Larger thermostat is rated at 500 watts, 110 or 220 volts a.c. for heater



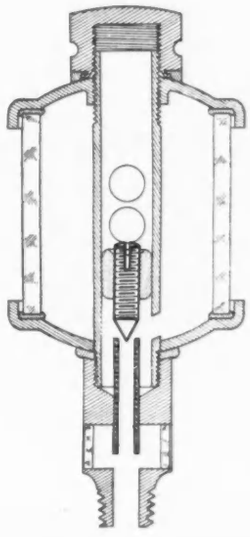
load. Body is 2¼ in. long and ⅝ in. in diameter, temperature differential is 5 deg. F. Either type supplied with contacts normally open or closed. Thermostats for operating temperatures up to 700 deg. F. available in the larger size. George Ulanet Co., 85 Columbia St., Newark, N. J.

Bi-Post Base Lamp

Increased durability, decreased weight and a more accurate positioning of the light source with relation to reflectors or lenses are claimed for the new 3,000-watt 32-volt airport floodlight lamp with a bi-post base, which is designated as a T-32 bulb. The bi-post base supersedes the GT-38 with prong base. The lamp has a light center length of 5¾ in. and an overall length of 14 in. Incandescent Lamp Dept., General Electric Co., Nela Park, Cleveland, Ohio.

Lubricator

Of air-tight construction, is self-contained and feeds oil under pressure generated by expansion of air within the lubricator as the bearing temperature rises. Design incorporates a large filler tube. Cadmium-plated steel is used for the body part for greater strength and to reduce possibility of breakage. Both reservoir supply and rate of feed are visible. Lubricator is said to be automatic, fool-proof and clean, preventing dripping or oil-throwing. Victor Lubricator Co., 3901 N. Rockwell St., Chicago, Ill.



MANUFACTURERS' PUBLICATIONS

Bearings — Norma-Hoffmann Bearings Co., Stamford, Conn. Catalog 940, 3½x11 in., 76 pages, "Norma-Hoffmann Precision Bearings—Engineering Data Sheets," includes sizes, dimensions, and load ratings for precision ball, roller and ball thrust bearings; typical applications, general mounting instructions given as well as complete catalog information.

Chain and Sprocket Wheels — Palmer-Bee Co., Detroit, Mich. Catalog Section 101, 7x10 in., 220 pages, includes catalog information and design data on a line of chain drives, truck wheels, conveyor chains and sprockets.

Contactors—Ward Leonard Electric Co., Mount Vernon, N. Y. Bulletin 4401, giving features and catalog information on a.c. magnetic contactors, and bulletin 1901 with similar information on d.c. magnetic contactors.

Controllers—Minneapolis-Honeywell Regulator Co., Minneapolis, Minn. Bulletin, 4 pages, "Automatic Pump Controller," describes features of construction and principles of operation. Another 4-page leaflet "Temperature Controllers," includes catalog and performance data.

Couplings — The Falk Corp., Milwaukee, Wis. Leaflet giving new dimensions and prices for the Falk-Rawson 4-duty coupling.

Drafting Equipment—Charles Bruning Co., Inc., 102 Reade St., New York, N. Y. Booklet, 14 pages, "Black and White Magic," describing the process and equipment used in a direct printing process.

Electric Motors—The Holtzer-Cabot Electric Co., 125 Amory St., Boston, Mass. Bulletin, 4 pages, giving design details and application for Type RWS split-phase motors in standard and special types.

Fiber—Wilmington Fibre Specialty Co., Wilmington, Del. Price list of sheets, rods and tubes.

Generators — General Electric Co., Schenectady, N. Y. Booklet, 4 pages, describing Type B, d.c. generators up to 5 kw. capacity.

Magnet Wire—Belden Manufacturing Co., Chicago, Ill. Folder, 9 pages, "Celenamel," describing enameled copper wire covered with cellophane; data includes dielectric properties, heat resistance, effect of solvents and comparative volume relations.

Monel Metal and Nickel—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. **Booklet, 4x7 in.,** "The Application of Monel Metal and Nickel to Industrial Processing Equipment," includes data on the physical and mechanical properties and an extensive list of applications classified according to material handled.

Motoreducers—The Falk Corp., Milwaukee, Wis. Bulletin No. 272, 36 pages, "Falk Motoreducer," describes constructional details, lubricating system and design of horizontal and vertical type of integral unit. Several pages devoted to the selection of types, service factors, rating tables and motor bases.

Motor Starters—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Folder L-20591 gives details of remote controls for constant and adjustable speed, d.c. motors.

Packings—Chicago Belting Co., 133 North Green St., Chicago, Ill. Booklet, 4 pages, includes an analysis of packing failures and several successful applications for hydraulic packings.

Precision Balance — Roller-Smith Co., 233 Broadway, New York. Catalog 240, 8 pages, describing a line of precision spring balances covering a range of 0.1 milligram to 50 grams.

Roller Bearings—The Bantam Ball Bearing Co., South Bend, Ind., catalog, 32 pages, 8½x11, "Bantam Straight Roller Radial and Thrust Bearings," lists speed factors, general modification factors, dimensions and capacities, and typical applications for a variety of types.

Screens—Link-Belt Co., 910 South Michigan Ave., Chicago, Ill. Booklet No. 1462, 16 pages, "Link-Belt Vibrating Screen," illustrating several designs.

Screen—Robins Conveying Belt Co., 15 Park Row, New York, N. Y., Bulletin No. 89, giving opening sizes, screen meshes and construction of woven wire cloth.

Steel—Union Drawn Steel Co., Massillon, Ohio. Two, four-page circulars, "Cold Work as a Substitute for Heat Treatment," and "Turned and Ground Shafting."

Strain Gage — Baldwin-Southwark Corp., Philadelphia, Pa. Bulletin No. 71, six pages, "Whittemore Strain Gage," describes reliability, applications.

Moments of Inertia of Rotating Bodies

(Continued from page 381)

eration of this useful effect of the flywheel and other rotating parts the amount of inertia should be under-estimated rather than over-estimated. But, in this same application, when it comes to estimating the power required to start from rest in a given time, or the capacity of controller required for starting, the moment of inertia should be over-estimated rather than under-estimated.

Now, having found out what a highly important part the moment of inertia plays in problems involving acceleration and having discussed ways for determining this value, there are certain practical points to be considered. The two factors making up this quantity are weight and radius. Since the latter enters into the problem as a squared quantity, increasing the radius has a greater effect on inertia than increasing the weight. Doubling the weight, for example, merely doubles the inertia, provided the radius remains the same, but doubling the radius, while the weight remains constant, increases the inertia four times. Hence, in designing high-speed parts to be started or reversed frequently, it is of first importance to keep their diameters small; next, to make them as light in weight as possible.

Even though high speed may not be involved, should the number of reversals or starts exceed, say, eight or ten per minute, these same factors should be carefully considered.

When we come to calculate torque and horsepower, we encounter still another factor—speed. It can be seen from Equations (3) and (4) that in determining the horsepower to accelerate a mechanism the inertia factor enters into the calculation only once, but the speed is there twice, or as a squared quantity. Thus, though the value of WR^2 should be kept low, it does not usually cause as much trouble as high speed. Often the importance of this point is overlooked by designers in laying out a drive which starts or reverses frequently. To have a mechanism reverse with the least expenditure of energy, the speed of its parts must be kept at the minimum.

Frequently motor manufacturers receive requests to supply reversing motors that are to run at 1800 r.p.m. It is far more desirable to use motors running at 900 r.p.m. or not more than 1200 r.p.m. The horsepower necessary to accelerate the 900-r.p.m. motor is only $\frac{1}{4}$ of that required for the 1800-r.p.m. motor plus what may be required to take care of the somewhat greater WR^2 of the slower-speed motor. Usually the gain from the reduced speed is much greater than the loss from increased WR^2. However, with the use of the slower speed, the external inertia, assuming no change, becomes a bigger percentage of the total. Actual calculations are frequently necessary to arrive at the proper combination of motor speed, and the speed of the external load.

In connection with this, reference to Equation (5) will bring out another point in regard to speed. When finding the equivalent inertia of rotating parts, the square of the speed also enters into the calculations.

Table II—Approximations for Calculating Moments of Inertia

NAME OF PART	MOMENT OF INERTIA
Fly Wheels (not applicable to belt pulleys)	Moment of inertia equal to 1.08 to 1.15 times that of rim alone
Fly Wheel (based on total weight and outside diameter)	Moment of inertia equal to 2/3 of that of total weight concentrated at the outer circumference
Spur or helical gears (teeth alone)	Moment of inertia of teeth equal to 40 per cent of that of a hollow cylinder of the limiting dimensions
Spur or helical gears (rim alone)	Figured as a hollow cylinder of same limiting dimensions
Spur or helical gears (total moment of inertia)	Equal to 1.25 times the sum of that of teeth plus rim
Spur or helical gears (with only weight and pitch diameter known)	Moment of inertia considered equal to 0.60 times the moment of inertia of the total weight concentrated at the pitch circle
Motor Armature (based on total weight and outside diameter)	Multiply outer radius of armature by following factors to obtain radius of gyration: Large slow speed motor.0.75 to 0.85 Medium Speed d.c. or induction motor.....0.70 to 0.80 Mill type motor.....0.60 to 0.65

Hence, if there is to be a gear train between the motor and the driven parts, it is advisable to make the reduction next to the motor as great as possible. In this way it is frequently possible to disregard the inertia of all parts beyond perhaps the first or second reduction. For example, with a 4:1 reduction adjacent to the motor and with no further reductions, anything beyond the motor shaft itself has only one-sixteenth the effect on the total inertia that it would have were it running at motor speed.

Importance of Light Weight

Of course, on rapid reversing jobs, all parts on the motor shaft, such as couplings or pinions, should be kept as small as possible. Too, the use of aluminum or aluminum alloys is often desirable for high-speed pulleys and sheaves. One example will illustrate. In a certain motor application the motor was burning up far too frequently. In checking up on the trouble, it was found that the motor had to start and stop many times each minute. Further, with only about a 2:1 reduction, next to the motor, the pinion was almost as large in diameter as the armature and was a solid disk. By increasing the reduction ratio and using a smaller, light-weight pinion, the difficulty was eliminated.

And now we consider what appears to be a paradox. In certain applications where motors overheat on rapid-reversing or frequent-starting cycles it is possible to remedy the trouble by putting in smaller motors. This is due to the fact that in very many cases the inertia of the motor rotor is the biggest factor in the acceleration load. In reversing planers, for example, the equivalent inertia of the parts external to the motor is frequently only 10 to 15% of the inertia of the driving motor. For this reason, a specification by a machine designer of the horsepower required for a reversing job, when he has not actually calculated the inertia, means very little. Unless the actual working portion of the cycle imposes a very great load, most of the work done will very likely be in reversing the motor. Hence to call for a 3-horsepower motor, just because it is bigger and, it would seem, ought to be better for the cycle, when two horsepower is enough, defeats its own purpose. In calling for a motor for a reversing cycle, it is, therefore, much better to specify the inertia of each part of the machine, or the equivalent inertia at the in-driving shaft, and the speeds, together with the loadings, over the cycle. Then, from this information it is possible to pick out the motor which has the best torque characteristics for the job.

In most cases the speed of the motor is higher than that of the rest of the mechanism. In some few instances the reverse is true, and the motor speed is less. In such an application a knowledge of the inertia of each part is of the utmost importance. Merely a 2:1 increase in speed means that the inertias of the higher-speed parts must be multiplied by four to place them on an equal basis with the motor. While in themselves they may not amount to much, such multiplications soon bring the external inertia up to, or in excess of, that of the motor. As a result, the power for accelerating both motor and load may become of some considerable magnitude.

Having found the horsepower required, having made whatever assumptions were necessary, and having taken all factors into consideration, it then becomes necessary to select the proper type of motor. The actual determination of horsepower can be made by the machine designer, but the selection of the type of motor is properly the function of some one familiar with motor characteristics and design.

The four principal types of squirrel-cage induction motors are as follows: (a) Normal torque, high starting current; (b) Normal torque, low starting current; (c) High torque, low starting current, low slip; (d) High torque, high slip, relatively low current.

The first two are considered general-purpose motors and usually are suitable for a limited number of reversals per minute or for not-too-frequent starting. When the starting service is severe, with heavy inertia loads to accelerate, type (*c*) is applicable. It also finds use for moderately frequent reversal when the running load between reversals is appreciable. The fourth, type (*d*), finds its widest use where heavy starting loads are encountered and where the running load is low, for instance on valves, or on moving tool heads or carriages of machine tools. It is also widely used on reversing applications where the number of reversals is very frequent, and where the major portion of the load is occasioned in reversing the motor rotor itself, with little actual running load. This type of motor, because of its high slip, is also admirably adapted to machines such as punch presses, for it can slow down under load and allow the flywheel to give up some of the kinetic energy.

Torque-Speed Curves

A comparison of the torque-speed curves of these types of motors is shown in Fig. 1. In general, the motor having the highest average torque throughout the accelerating period—though not necessarily the highest torque—is the one best suited for applications requiring frequent acceleration. This is based on the assumption that the time for accelerating the load is limited. In fact it usually is limited since, when frequent acceleration is necessary, time becomes an important factor. For this reason the motor which has the highest *average* torque during acceleration and therefore requires the least time to bring the load up to speed, is the one most generally applied. Other motors, having lower average torques during accelera-

tion, might also be capable of bringing the load up to speed, but none of them could do so as rapidly.

In determining the available torque of the motor when the actual torque-speed curves are not at hand, it is customary to use an average of the starting, maximum, and full-load running torques of that motor. These as a rule are given in the motor data. In the case of motors of types (*c*) and (*d*), previously mentioned, it is found sometimes that only the starting and the full-load torques are given. However, the average of these two values can usually be used safely in calculations, for it is generally less than the actual average.

The torque calculations, as previously mentioned, make no provision for a varying value of effective accelerating torque, but are based on the assumption that the torque—hence the angular acceleration—remains constant throughout the acceleration period. To be exact, corrections should be made to compensate for the variation in torque. For example, the calculations might be made by small steps for each increment of velocity with respect to time in Equation (3) as the machine is coming up to speed—on the theory that if the steps taken are small enough the acceleration will be practically constant during each step. Or it might be possible to determine the progressive rate of change of velocity in relation to time during the accelerating period and to obtain a more accurate solution of Equation (3) by the application of calculus.

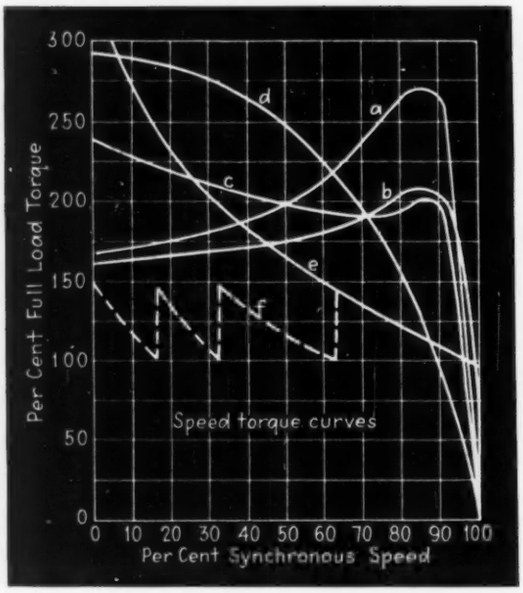


Fig. 1—Speed Torque Curves of Typical Motors
- **(a) Normal Torque, High Starting Current.**
- **(b) Normal Torque, Low Starting Current.**
- **(c) High Torque, Low Starting Current, Low Slip.**
- **(d) High Torque, High Slip with relatively Low Starting Current.**
- **(e) D.C. Motor, Full Voltage.**
- **(f) D.C. Motor, Resistor Step Starter.**

However, in view of the inexactness of the data usually available and the variability of conditions, it usually is sufficiently accurate to base the calculations on the average torque without any correction, as previously discussed in connection with Fig. 1.

An additional curve is shown on Fig. 1. This is for a direct-current motor with a starting resistor. With this equipment the torque can, within limits, be kept nearly constant by properly proportioning the design of the resistor and starter. As shown by the curve, the starter in this case is designed to limit the starting current and the current of each successive peak to approximately 150 per cent of full-load current, and to cut out the next block of resistance at about normal full-load current. By altering the design of the starter, it is possible to vary the height of the current peaks and, correspondingly, of the torque peaks between any values within the commutating limits of the motor.

Selection of the Motor

From the foregoing discussion, it can be seen that, although selection of the particular type of motor to be used is not properly within the province of the mechanical engineer, its size is governed entirely by his designs. He can, by a little calculation, readily determine just what horsepower will be required to provide the rate of acceleration he desires. In fact, he is the only one who can do this, unless he is willing and able to give the motor man more complete details of the application than are usually made available with a request for motor recommendations.

This last point frequently stands as the motor man's greatest stumbling block in his efforts to apply the most suitable motor to the job. Too often, the information provided by the person requesting motor-application advice is so scant and so lacking in detail that any recommendations made under the circumstances can, at the best, be only approximations.

The remedy, of course, lies in the hands of the mechanical engineer. He need only make the simple calculations, outlined above, and include with his request for recommendations the answer to the question asked in the first paragraph: "What is the external inertia, or WR^2, referred to the motor shaft?"